EXPERIMENTAL STUDY OF IMPINGEMENT COOLING
IN ROTATING TURBINE BLADES

by

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Submitted to the Department of Aeronautics and Astronautics
in Partial Fulfillment of the Requirements of the Degree of

DOCTOR OF PHILOSOPHY IN GAS TURBINES

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

September 1983

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NOV 30 1983 Archives
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ABSTRACT

The objective of the present work is the investigation
of the effects of rotation on impingement and internal heat
transfer in turbine blades.

Results from the first set of parametric experiments are
presented. They provide thermal measurements on a blade
model, which are finely resolved in space, so that variations
due to individual impinging jets can be observed. They also
demonstrate that rotation can generate hot spots and cause a
substantial, speed dependent reduction of the average heat
transfer coefficient. Nusselt number maps and static data
correction factors, applicable to rotor blade cooling design,
are included. A qualitative theoretical explanation of the
observed phenomena is given. A comparison with the results of
other investigators is made and conclusions are drawn.

A detailed description of the M.I.T. - G.T.L.
Impingement Cooling Facility, conceived, designed and built
for this research is given. The experimental concept utilizes
a large scale, electrically heated, thin walled, impingement cooled model of a turbine blade which is rotated in a vacuum chamber. Its external wall temperature is measured with high spatial resolution by an infrared radiometer. Knowledge of the temperature distribution and the electrical heating power generated in the wall allows the calculation of the local heat transfer coefficient. Additional temperature and pressure measurements are made inside the blade model cooling gas channels.

The similarity considerations which guided the design of the experiment are discussed. All important non-dimensional test parameters, including Reynolds, Rossby, Mach and Prandtl numbers as well as temperature and geometrical magnification ratios were chosen to simulate actual operating conditions in gas turbines.

Suggestions are given for future research using the M.I.T. - G.T.L. Impingement Cooling Facility.

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AKNOWLEDGMENTS

This research was funded by Teledyne CAE under the Agreement of 4-23-79, which also undertook the design and manufacture of the vacuum chamber and shaft. Additional funding was provided by NASA Lewis Center under Grants NGL 22-009-383 and NAG 3-335. Their support is acknowledged with thanks.

Professor Jack L. Kerrebrock put a lot of enthusiasm, hard work and funds into this research. His advice, endless encouragement and personal warmth make me feel honored and privileged to have worked under him.

Professor Alan H. Epstein, with constant discussions, suggestions and critical comments, provided invaluable assistance to my understanding of instrumentation and data acquisition. I feel indebted to him for handling the finances of the project during the last two years.

Professor William T. Thompkins' stimulating questions have helped me greatly in the data reduction and analysis.

Further, I would like to express my sincere thanks to:

Mr. Bob Haines for developing the computer / instrumentation interface subroutines.

Messrs. Bob H. Bush, Jeff R. Gertz, Gerry R. Guenette and Kenneth C. Hall for their help during the software development and the many elucidating discussions.

Mr. Uriel Z. Preiser for his assistance in conducting the experiments.
The technical and support personnel of the M.I.T. Gas Turbine Laboratory for putting the facility together and providing expeditiously everything necessary to build it and keep it running. Special thanks to Mr. John N. Stanley for the advice, patience and skill he put into the instrumentation and assembly of the blade model, and for his help in conducting the first crucial experiments.

Finally, I would like to express my gratitude to my beloved wife Katerina and our families for their love, encouragement, patience and support, and dedicate this work to them.
NOMENCLATURE

GREEK SYMBOLS

\( \beta \) = \( 1/T \) - gas coefficient of expansion (1/oK)
\( \gamma \) = \( C_p/C_v \) - specific heat ratio (rad)
\( \gamma \) = stagger angle (rad)
\( \varepsilon \) = emissivity (oK)
\( \Delta T \) = \( T - T_c \) - temperature difference (oK)
\( \Delta T \) = \( T_{w} - T_{c} \) - reference temperature difference (oK)
\( \theta \) = shaft phase angle (rad)
\( \mu \) = viscosity (N sec/sq.m)
\( \rho \) = density (kg/cu.m)
\( \sigma \) = 5.67E-8 = Stefan - Boltzmann constant (W/sq.m oK)
\( \phi \) = dissipation function (l/sec)
\( \Omega \) = angular velocity (rad/sec)

ENGLISH SYMBOLS

a = slope of the model surface (rad)
Af = surface area of resistive wall (sq. m)
B = body force (N)
Ba = body force due to axis motion (N)
Bb = body force due to buoyancy (N)
Bg = body force due to gravity (N)
Cp = specific heat under constant pressure (J/kg oK)
CF = correction factor (1)
D = jet diameter (reference length) (m)
Dd = leading edge diameter (m)
D# = distance to the center of adjacent spot # (m)
D/Dt = substantial derivative (m/s)
E = \( V/(C_p\Delta T) = (\gamma - 1)M T/\Delta T \) = Eckert number
fo = translational acceleration (m/s^2)
G = \( d/d' \) = geometrical magnification ratio (1)
h = heat transfer coefficient (W/sq.m oK)
H = enthalpy (J/kg)
i = serial number of neighboring spot (A)
I = current (A)
k = thermal conductivity (W/m oK)
M = \( V/\sqrt{\gamma RT} \) = Mach number
Mj = \( V_j/\sqrt{RT} \) = jet Mach number
MM = molecular mass (kg/mole)
N = 12 = number of jets (N)
Nu = \( h d/k \) = Nusselt number (N)
P = pressure (N/sq.m)
Pc = jet hole pitch (m)
Pc = coolant supply pressure (N/sq.m)
Pr = \( \mu C_p/k \) = Prandtl number (N)
P# = pressure at port #, normalized by Pc (N)
Q = power (W)
r = radius (m)
r = position vector (m)
R = gas constant (J/kg oK)
Ra = Rayleigh number (W)
R = Rayleigh number (J/kg oK)
Re = \frac{4M}{\mu N \pi d} = \text{average jet Reynolds number} \quad \text{(Ohm)}

Rf = \text{foil resistance} \quad \text{(m)}

Rv = \text{hub radius} \quad \text{(m)}

Ro = \frac{V}{d} = \text{Rossby number} \quad \text{(m)}

Rt = \text{tip radius} \quad \text{(m)}

S = \text{span} \quad \text{(m)}

Si = \text{impingement cooled span} \quad \text{(m)}

S# = \text{length of side # of the finite element} \quad \text{(m)}

t = \text{time} \quad \text{(sec)}

T = \text{finite element (wall) thickness} \quad \text{(m)}

T = \text{temperature} \quad \text{(oK)}

Ta = \text{ambient temperature} \quad \text{(oK)}

Tc = \text{coolerant supply temperature (reference temperature)} \quad \text{(oK)}

Tj = \text{temperature at jet exit} \quad \text{(oK)}

To = \text{spot temperature} \quad \text{(oK)}

Tw = \text{average wall temperature} \quad \text{(oK)}

T# = T at the center of neighboring element # \quad \text{(oK)}

u = \text{x-velocity component, velocity vector} \quad \text{(m/sec)}

v = \text{y-velocity component} \quad \text{(m/sec)}

V = \text{jet velocity (reference velocity)} \quad \text{(m/sec)}

W = \text{z-velocity component} \quad \text{(m/sec)}

W = \text{power generated per unit of fluid volume} \quad \text{(W/cu.m)}

X = \text{x-coordinate on the unfolded model skin} \quad \text{(m)}

Y = \text{y-coordinate on the unfolded model skin} \quad \text{(m)}

z = \text{wall to jet distance} \quad \text{(m)}
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CHAPTER 1

INTRODUCTION

1.1 Introduction.

During the past two decades high pressure turbines of aircraft engines have been operating in an increasingly hostile environment. Temperature, pressure and rotational speed levels kept increasing and this trend will certainly continue for the years to come, in our effort to increase both efficiency and thrust to weight ratio.

Although blade metallurgy has made some spectacular advances during the same period, the main factor which currently determines the lifetime of a high pressure turbine is cooling. The blade metal temperature must be held several hundred degrees below the gas temperature to control stress rupture, creep and, to some extent, corrosion / erosion on a highly local basis, in both steady state operation and during rapid transients, and without excessive thermal stresses. This must be accomplished with the minimum cooling air flow in order not to degrade the overall efficiency of the engine.

These requirements are especially important for rotor blades not only because of the very high centrifugal stresses imposed on them, but also because of the severe consequences of a rotor blade failure (blade ejection or digestion, shaft unbalance, etc.)

Impingement cooling (Fig.13) has long been recognized as
a highly effective method of internal blade cooling, providing heat transfer coefficients comparable to those encountered in nucleate boiling of water and, consequently, has been widely applied in aircraft engines.

The leading edge of a turbine blade in particular is a region where the hot gas side heat transfer coefficient is very high (Ref. [19]) because of the existence of a stagnation point. Impingement cooling, used alone or in combination with film cooling, provides a means of introducing stagnation points on the cooling gas side. There is a large amount of data in the open literature, both theoretical and experimental (Ref. [1] to [15]) accounting for different geometries and configurations. Because of the complicated geometry involved in the leading edge impingement cooling and its aforementioned importance for practical applications, experimental parametric studies have been conducted (Ref. [1] to [7]) which resulted in empirical correlations or graphs applicable to the design of turbine vane cooling.

1.2 Problem Identification.

Two facts should be considered however:

i) All the impingement cooling experiments mentioned above were done in a fixed (or laboratory) coordinate system, simulating thus stator vanes but not necessarily rotor blades.

ii) Published data give the spanwise averaged heat transfer coefficient on the leading edge, providing spatial resolution only in the chordwise direction.
Gas turbines operate inherently at very high rotational speeds, which introduce centrifugal, Coriolis and buoyancy forces on the cooling gas inside the rotor blades. These forces couple the momentum with the energy equation (See Sect. 2.1,) making the already difficult problem of solving the governing differential equations of motion even more so. Many researchers have conducted experimental and theoretical studies to investigate rotational effects. Their results could be summarized as follows: Rotation:

i) Gives rise to secondary flows (Ref.[27], [28])

ii) Affects the stability of boundary layers (Ref. [29], [30], [31], [32])

iii) Causes changes in the heat transfer process (Ref. [22], [23], [24], [25], [26].)

In the absence of any data on the effects of rotation on impingement heat transfer, the assumption in the typical cooling design system (Ref.[17]) is that there is no effect.

In order to predict the lifetime of a cooled blade, the temperature distribution inside its wall is computed by numerically solving the Poisson heat conduction equation. The solution and the accuracy of the prediction depends exclusively on the thermal boundary conditions prescribed on the external and internal wall surfaces. It is therefore very important to have accurate and spatially finely resolved heat transfer data during the design process.

The present work is mainly concerned with the rotational effects on turbine blade impingement cooling. In addition, it
provides highly resolved measurements of the local heat transfer coefficient distribution, under rigorously simulated rotor operating conditions.

1. 3 Preliminary Experimental Considerations.

It would be highly desirable to measure local heat transfer coefficients on a turbine rotor blade under actual operating conditions. The hostile environment makes this extremely difficult, if not impossible, to achieve with current instrumentation technology. Therefore measurements under closely simulated operating conditions are an attractive and viable alternative.

Heat transfer associated with blade cooling consists of:

i) Convection from hot gas to outer blade wall.

ii) Conduction through the blade wall.

iii) Convection from inner blade wall to cooling fluid.

Determination of the local heat transfer coefficient distribution on either side requires highly resolved measurements of the temperature distribution on both wall sides. This requirement for high spatial resolution dictates either:

i) Dense instrumentation with thermocouples and/or heat transfer gauges, which can introduce distortions in the temperature and flow fields. Transducer signals must come out of the rotating system through a large number of instrumentation slip rings or telemetry channels. Or,

ii) Use of a non-contact radiometric method, which is
limited to the measurement of the external blade surface temperature distribution. For real turbine blades, this information alone, although useful for other purposes, is not adequate for the determination of the local heat transfer coefficient, because of the unknown internal wall temperature.

In order to profit from the advantages and overcome the above mentioned and other drawbacks (such as hot gas and particle interference, obscuration by adjacent blades, reflection errors introduced by flame radiation and neighboring hot parts) associated with non-contact temperature measurement under actual operating conditions, an experimental facility, shown in Fig. 1 and 17 was conceived, designed and built at the M.I.T. Gas Turbine Laboratory. Its purpose is to isolate the convection process from inner blade wall to cooling fluid and to allow quantitative study of heat transfer phenomena occurring in internally cooled turbine blades under rotation.

The experimental concept is as follows: A thin stainless steel foil is heated by uniform resistive dissipation, while being cooled from one side by an air jet and observed from the opposite side (Fig. 14.) Temperature gradients in the directions parallel with the foil surface, produced by the cooling jet reach the observer's side virtually unaltered. If the observer's side is thermally insulated, the jet side heat transfer coefficient can be computed from the measured temperature distribution and the known power input.
The fundamental theoretical aspects of the scaling rules, the implementation of the experimental concept and the unique radiometric measurement technique used are presented more rigorously in the following chapters. Descriptions of the experimental facility and its subsystems along with some of the problems encountered which dictated major design choices can be found in Ref. [16] and are presented here as well, in an updated form, for completeness.
CHAPTER 2

SIMULATION

2.1 Equations of Motion in Rotating Coordinates.

The system of coordinates is shown in Fig.15, rotating with constant angular velocity about the Z-axis. It is chosen so that the turbine blade is stationary with respect to it.

The cooling fluid is considered to be a perfect gas, of constant viscosity, specific heats and thermal conductivity. The governing equations of motion are in general:

Mass Conservation:

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \quad (\text{Equ. 2.1}) \]

Momentum Conservation:

\[ \frac{D \mathbf{u}}{Dt} = - \nabla P + \mu \left[ \nabla (\nabla \cdot \mathbf{u}) + \frac{2}{3} \nabla \mathbf{u} \right] + \rho \mathbf{B} \quad (\text{Equ. 2.2}) \]

Energy Conservation:

\[ \frac{D H}{Dt} = k \nabla^2 T + \frac{D p}{Dt} + \mu \phi + W + \rho \left( \mathbf{B} \cdot \mathbf{u} \right) \quad (\text{Equ. 2.3}) \]

State Equation:

\[ p = \rho R T \quad (\text{Equ. 2.4}) \]

The body force \( \mathbf{B} \) has three components. The first, \( \mathbf{B}_a \), is due to the axis motion, the second, \( \mathbf{B}_b \), is due to buoyancy, and the third, \( \mathbf{B}_g \), is due to the gravitational field of the earth. The latter may be neglected as very small compared
with the others.

The force due to axis motion is (Ref.[21]) in general:

\[ \dot{\vec{B}}a = - \dot{\vec{f}}_o - 2 \vec{\Omega} \times \vec{\dot{u}} - \frac{d\vec{\Omega}}{dt} \times \vec{r} - \vec{\dot{\Omega}} \times (\vec{\Omega} \times \vec{r}) \]  

(Equ.2.5)

In the case under study there is neither translational acceleration nor time dependence, since steady state turbine operation is assumed. Therefore \( \dot{\vec{B}} \) simplifies to:

\[ \dot{\vec{B}}a = - 2 \vec{\Omega} \times \vec{\dot{u}} - \vec{\dot{\Omega}} \times (\vec{\Omega} \times \vec{r}) \]  

(Equ.2.6)

The force due to buoyancy is:

\[ \dot{\vec{B}}b = - \dot{\vec{B}}a \beta \theta \]  

(Equ.2.7)

Taking in account Equ.2.6 and 2.7 and also that:

Steady State: \[ \frac{\partial}{\partial t} = 0 \]  

(Equ.2.8)

No Heat Sources in the Fluid: \[ W = 0 \]  

(Equ.2.9)

Perfect Gas: \[ \beta = \frac{1}{T} \]  

(Equ.2.10)

and introducing appropriate reference quantities, Equ.2.1 through 2.3 can be written in a simplified, non-dimensional form.

Mass Conservation:

\[ \nabla' \cdot (\rho' \vec{\dot{u}}') = 0 \]  

(Equ.2.11)

Momentum Conservation:

\[ \rho' (\vec{\dot{u}}' \cdot \nabla') \vec{\dot{u}}' = - \nabla' p' + \frac{\mu}{\rho' \cdot d} \left[ \nabla' \left( \nabla' \cdot \vec{\dot{u}}' \right) \right] + \nabla^2 \vec{\dot{u}}' + \left( \vec{\dot{\Omega}} \frac{d}{V} \left( -2 \vec{\dot{\Omega}} \times \vec{\dot{u}}' \right) \right) \left[ \frac{\Omega \cdot d}{V} \right]^2 \left( -\vec{\dot{\Omega}} \times (\vec{\dot{\Omega}} \times \vec{\dot{r}}') \right) \rho' \left( 1 + \beta' \theta' \right) \]
Energy Conservation:  
\[ \rho' (u' \cdot \nabla') H' = \frac{\mu}{V \rho d} \frac{k}{C_p \mu} \frac{\nabla^2}{T} + \frac{V}{C_p \Delta T} (u' \cdot \nabla') \rho' + \]
\[ \frac{\mu}{V \rho d} \frac{\nabla^2}{C_p \Delta T} \phi + \left[ \frac{\Omega d}{V} \right]^2 \frac{V}{C_p \Delta T} \left[ -\nabla' \times (\Omega' \times \nabla') \right] \rho' (1 + \beta' \Theta') \]

2.2 Similarity Conditions.

This system of non-linear, partial differential equations is governed by the following four independent dimensionless parameters:

Reynolds number: \( Re = \frac{V \rho d}{\mu} \)  
(Equ.2.14)

Prandtl number: \( Pr = \frac{C_p \mu}{k} \)  
(Equ.2.15)

Eckert number: \( E = \frac{V}{C_p \Delta T} = (\gamma - 1) \frac{M^2}{\Delta T} \)  
(Equ.2.16)

Rossby number: \( Ro = \frac{V}{\Omega d} \)  
(Equ.2.17)

The particular solution and consequently the dimensionless local heat transfer coefficient known as Nusselt number (Nu) corresponding to the physical problem of internal rotor blade cooling is determined by these quantities and the dimensionless boundary conditions imposed in each case.

It should be noted here that a non-dimensional parameter associated with natural convection and buoyancy effects, known as Rayleigh number (Ra) can be formed from the
above:

\[ \text{Ra} = f(\text{Re}, \text{Ro}, \text{Pr}, \frac{\Delta T}{T}) = \frac{r}{d}\frac{\text{Re}^2 \text{Pr}(\Delta T/T)}{\text{Ro}} \quad (\text{Equ.2.18}) \]

2.3 Implications of the Similarity Conditions.

Let a prime denote test conditions for the rest of this chapter. Similarity of the boundary conditions requires that the laboratory model be similar in a geometrical sense with the actual turbine blade.

Temperature similarity requires equality of dimensionless temperature gradients or temperatures between the model and the actual blade on all boundary surfaces, i.e. the ratios of all temperatures to some reference temperature, and the spatial distributions of these ratios, must be identical between model and blade. To begin with, the ratio of the mean wall temperature to coolant temperature is kept the same, since it is the driving factor for buoyancy effects, as Equ.2.12 suggests.

Reynolds number similarity implies that:

\[ \text{Re} = \frac{V \rho d}{\mu} = \frac{4M}{N \mu \pi d} = \text{Re}' \quad (\text{Equ.2.19}) \]

Prandtl number depends mainly on the cooling fluid and to a lesser degree on temperature and pressure. Since a gas is used as the coolant in both real and test conditions, it is reasonable to assume for the moment - and verify later - that:

\[ \text{Pr} = \text{Pr}' \quad (\text{Equ.2.20}) \]
Eckert number equality implies that:

$$E = \frac{V^2}{C_p \Delta T} = (\gamma - 1) \frac{M^2 T_j}{\Delta T} = (\gamma' - 1) M^2 \frac{T_j'}{\Delta T'} = E' \quad \text{(Equ. 2.21)}$$

Mach number is related to disturbance propagation and must be kept the same. As noted above (Equ. 2.13), the temperature ratio is the driving factor for buoyancy effects and, therefore, must also be kept the same. This means that, for Eckert number equality, the specific heat ratios of real and test cooling gas should be equal. This condition will be qualified as noted below:

Mach number equality implies that:

$$M = \sqrt{\frac{V}{\gamma R T_j}} = \sqrt{\frac{V'}{\gamma' R' T_j'}} \implies V' = V \sqrt{\frac{\gamma' MM T_j'}{\gamma MM' T_j}} \quad \text{(Equ. 2.22)}$$

or:

$$V' = V \sqrt{\frac{\gamma' MM T c'}{\gamma MM' T c}} \quad \text{(Equ. 2.23)}$$

and finally:

$$V' \sim V \sqrt{\frac{\gamma' MM T c'}{\gamma MM' T c}} \quad \text{(Equ. 2.24)}$$

Rossby number similarity implies that:

$$Ro = \frac{V}{\Omega d} = \frac{V'}{\Omega' d'} \implies \Omega' = \Omega \frac{d}{d'} \frac{V'}{V} \quad \text{(Equ. 2.25)}$$

and by virtue of Equ. 2.24:

$$\Omega' = \Omega G \sqrt{\frac{\gamma' MM T c'}{\gamma MM' T c}} \quad \text{(Equ. 2.26)}$$

where:

$$G = \frac{d}{d'} \quad \text{(Equ. 2.27)}$$

It is desirable to conduct the experiment at low
tangential velocity, in order to reduce centrifugal stresses
on the rotor, simplify its design and alleviate some of the
other problems associated with high speeds (bearings, seals
etc.) Equ.2.26 suggests that, in order to achieve this one
should:

i) Use a large scale model,

ii) Use a cooling gas of high molecular mass,

iii) Conduct the experiment at low absolute temperature,

iv) Use a cooling gas of low specific heat ratio.

Requirement (iii) relaxes thermal stresses on rotating
parts. In addition, low enough temperature allows use of
materials such as high temperature elastomers and epoxy resins
and other insulators and sealants, which greatly simplify the
blade model design and manufacture.

Requirement (iv) however contradicts the last of the
stated conditions for Eckert number similarity. Its modest
(10%) speed reduction savings could be sacrificed for full
similarity. A mixture of monatomic - polyatomic gases such as
Argon - Refrigerant-12 or Xenon - Refrigerant-14 could provide
a specific heat ratio equal to 1.3, while keeping the
molecular mass high enough for a substantial speed reduction.
Several considerations however dictated the use of a
condensible fluid.

Environmental considerations, and cost per experiment,
suggested a closed loop cooling system. The sensitivity of
metal - to - gas heat transfer to oil contamination and the
possibility of oil decomposiition or even coking at blade
model temperature dictated the use of a clean, oil-free compression system. An oil-free compressor could allow the use of gas mixtures such as the ones mentioned in the previous paragraph. The high cost of such equipment however imposed the choice of a condenser and liquid pump system, shown in Fig.28, using Refrigerant-12 (R-12 or dichlorodifluoromethane) with a specific heat ratio of 1.1 as the cooling gas (see Sect.3.6.)
CHAPTER 3

THE EXPERIMENTAL APPARATUS

3.1 Fundamental Experimental Concept.

An illustration of the fundamental experimental concept was given in Sect.1.3 with the aid of Fig.14. A more rigorous examination follows:

Consider a finite element of the model wall shown in Fig.16, which is heated electrically by uniform resistive dissipation. Let $T_i$ be the temperature at the center of the finite element and $T_{i+1}$ the temperature at the centers of its neighboring elements. Although temperature gradients in the direction normal to the wall are large, due to the intense cooling, temperature differences between the outer and inner surfaces are negligible (less than 0.2 oK) because of the thin (0.1 mm) wall. So, for this analysis, the wall is considered isothermal in the normal direction.

The chamber vacuum reduces convective heat transfer from the outer wall surface to less than 0.1% of the total power input, which is neglected, and only losses by radiation are considered. At steady state the power balance of the finite element yields:

$$\dot{Q}_{\text{select}} + \dot{Q}_{\text{conv}} + \dot{Q}_{\text{cond}} + \dot{Q}_{\text{rad}} = 0$$  \hspace{1cm} (Equ.3.1)

where:
\[
\dot{Q}_{\text{select}} = \frac{2}{I_Rf} \frac{I_Rf}{S_1 S_2 t} = \frac{2}{I_Rf} \frac{I_Rf}{S_1 S_2} \quad \text{(Equ. 3.2)}
\]

\[
\dot{Q}_{\text{conv}} = h S_1 S_2 (T_c - T_o) \quad \text{(Equ. 3.3)}
\]

\[
\dot{Q}_{\text{cond}} = \sum_{i=1}^{4} \frac{k}{D_i} (T_i - T_o) \quad \text{(Equ. 3.4)}
\]

\[
\dot{Q}_{\text{rad}} = \varepsilon \sigma S_1 S_2 (T_a - T_o)^4 \quad \text{(Equ. 3.5)}
\]

Substituting the above values and solving Equ. 3.1 for \( h \) yields:

\[
h = \frac{\frac{2}{I_Rf} \frac{k}{S_1 S_2} \sum_{i=1}^{4} \frac{S_i (T_i - T_o)}{D_i}}{T_o - T_c} + \varepsilon \sigma (T_a - T_o)^4 \quad \text{(Equ. 3.6)}
\]

Heat transfer in gas turbine blades is three-dimensional, and heat is conducted through and not generated in the blade material. For the internal heat transfer process however, what matters is the temperature at and the heat flux through the inner wall surface. A thin wall, which simulates these boundary conditions, allows observation of the traces of the internal cooling process from the outside and deduction of quantitative results about the heat transfer coefficient distribution.

3. 2 Vacuum Chamber and System.

The insulation of the outside of the model wall was implemented in a way which allows non-contact temperature measurement. The blade is rotated in a vacuum chamber (Fig. 17.) The vacuum limits heat transfer on the outside of
the blade to conduction through the supporting structure and radiation, which is small at the temperatures of interest. Additional advantages of this concept are small power requirement for the driving motor and elimination of any absorbing medium between radiometer and measurement spot.

A 2.2 kW, 38 lt/sec (3 hp, 80 cfm) vacuum pump (STOKES model 149H-11) pumps the chamber to a vacuum of 0.3 mm Hg. A filter is used to minimize oil backstreaming that could cause contamination of optical surfaces. Access to the inside is achieved by means of a wheeled cover and, under vacuum, by means of electrical, hydraulic, and mechanical feedthroughs.

Three sets of 50.8 mm (2 in.) diameter, double, carbon-ceramic end face seals (JOHN CRANE type 21,) modified for pressure balancing, are used. They serve three functions:

i) Provide a rotational motion feedthrough for the shaft while maintaining the vacuum inside the chamber.

ii) Positively separate the intake and exhaust streams of coolant.

iii) Prevent any leakage of coolant to the atmosphere and vice versa.

Refrigerant-11 (R-11, trichloromonofluoromethane) is used in a pressurized, closed loop to cool and lubricate the seals. Despite its low boiling point and its poor lubrication qualities, its use was deemed necessary because, if any leakage is mixed with the cooling gas (R-12,) R-11:

1) Does not alter the molecular mass of the cooling gas significantly.
ii) Does not contaminate the surfaces. An oily film on the inner skin surface would significantly alter heat transfer in the blade model. Oil vapour could contaminate optical surfaces inside the vacuum chamber.

iii) Does not freeze at Refrigerant-12 condensation or subcooling temperatures (See Sect. 3.6, 3.7)

3.3 Rotor.

The rotor, shown in Fig. 2, 3 and 17, consists of the shaft, blade model, calibrating body, supporting structure and shaft mounted heat exchangers, instrumentation and current supply system components.

The shaft is made of 17-4PH stainless steel and is shown in Fig. 3 and Fig.17. It rotates on two spring loaded deep groove ball bearings (MRC 312 SF,) and serves three main functions:

i) Rotates the blade model, calibrating body and their supporting structure inside the vacuum chamber, driven by a 2.2 kW (3 hp) DC motor (MARATHON ELECTRIC model DL 781) via a single V-belt. A 1000 Ampere welding rectifier (AIRCO model 7.5-3.75-1.875 DDR-24-C7) is used as the power supply for the driving motor. No speed control is used. Narrow range speed adjustment is achieved by varying the output voltage of the rectifier and/or the armature and/or field resistance of the motor with external rheostats. Large speed changes require alteration of the transmission ratio by means of the driving sheave.
ii) Provides passages for the intake and exhaust of the blade coolant (two 15.9 mm (0.625 in) diameter off-axis bores.)

iii) Provides a conduit for 16 instrumentation leads (one on-axis 7.9 mm (0.310 in) diameter bore.)

The blade model is a four (4) times scaled up version of the leading edge geometry of a real turbine blade of interest.

The configuration used for all the tests included here is shown in Fig. 7 and 18. The test section geometry is shown in Fig. 19 and its dimensions are given in the following table:

<table>
<thead>
<tr>
<th>TABLE # 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>TEST SECTION GEOMETRY</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Span</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Hub radius</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Tip radius</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Leading edge diameter</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Stagger angle (from axial)</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Impingement hole diameter</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Impingement hole pitch</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Wall to jet distance</td>
</tr>
<tr>
<td>----------------------------</td>
</tr>
<tr>
<td>Impingement insert span</td>
</tr>
</tbody>
</table>

As can be seen from Fig. 18 and Table # 1, the jet array cools only 3/4 of the skin span. Although this configuration does not simulate the full scale blade geometry, it was necessary in order to accommodate the coolant exhaust and to allow mounting of the thermocouple at the impingement space tip. The model design was influenced by the following
requirements:

i) Adaptability, so that the same hardware can simulate as many different impingement cooling configurations as possible with only minor modifications and minimum facility downtime.

ii) Ease of manufacture, instrumentation and final assembly.

To better understand how these two requirements were satisfied, a closer look at the details of the design and components is necessary.

A fixed flange (Fig.11, 18) supports the model structure. Two circular sector shaped cavities form the intake and exhaust plenums. A third, kidney shaped cavity supports an insulated copper electrical contact. The shape of the cavities allows rotation of the moving flange, i.e. change of the stagger angle, up to 60 degrees from the axial direction by 15 degree increments with no modification of the electrical or the gas supply and exhaust lines. An O-ring seal prevents loss of coolant into the vacuum chamber. A flat silicone rubber gasket positively separates the intake and exhaust streams.

A center hole is provided for the instrumentation strut which will be described in detail in Sect.4. 2.

The moving flange is the base for the impingement insert, and is fastened to the fixed flange at a chosen stagger angle.

The impingement insert consists of the following parts:

i) The jet array, necessary to simulate the length to diameter ratio of the actual impingement jet holes.
Modifications to the jet array allow variation of parameters such as hole diameter, pitch/diameter and/or hole length/diameter ratios.

ii) The instrumentation spine, used to support the two instrumentation strips. These are dovetail-shaped, stainless steel strips carrying thermocouples, pressure taps and the associated wires and tubes. After complete assembly of the instrumentation outside the model, the strips are inserted in similarly shaped slots in the instrumentation spine, with the wires and tubes accommodated in grooves provided for this purpose.

iii) Top and side covers which form and seal the impingement supply plenum.

The cover (Fig. 7 and 18) provides a structural, electrically insulating support for the thin resistive skin, electrical terminals and conductor, and seals the cooling gas from the vacuum. By design, the cover can slide in a shallow, wide groove, allowing thus variation of the z/d ratio. The electrical conductor has a slot parallel to the groove to accommodate this motion. Once at the desired position, the cover is held in place and supported against centrifugal forces by two clamps. A flat, silicone rubber gasket is used as a vacuum seal.

The thin resistive skin of the blade model is made of Kanthal A-1 (registered trademark of Kanthal Corporation,) a special alloy with high resistivity, 1.45E-06 Ohm m (872 Ohm/cmil ft,) low thermal conductivity, 16.74 W/m oK (9.672
Btu/hr ft °F) and very low temperature coefficient of resistivity (+/- 50 ppm/oK,) properties which make it the optimal choice for this application. Two strips, 32.5 mm (1.280 in.) wide, were selected for uniform thickness from a 97 μm (3.8 mils) thick foil, degreased, sandblasted with 50 μm grit alumina and oxidized for 8 hrs at 1366 oK (2400 °F) in air, in order to increase their surface emissivity to approximately 0.7 (not measured). Copper terminals were brazed at both ends of one strip, which is folded to form a semi - cylinder of 12.7 mm (0.5 in.) diameter, and the assembly is bonded on the cover with high temperature epoxy adhesive.

The other strip is bonded on the outer surface of a copper tube to form the calibrating body, shown in Fig. 8. A cladded electrical heater is brazed on the inner tube surface to allow temperature level adjustment by variation of the current input. Two thermocouples, mounted at the tip and hub of the calibrating body, measure the temperature at both ends.

The calibrating body provides a quasi - isothermal surface with the same geometry and emissivity as the blade model. Thus, the radiometer can be calibrated and automatically compensated for emissivity and geometry effects.

The outer diameter of the calibrating body surface is equal to the leading edge diameter of the blade model. In order to follow the blade model when parametric geometry changes occur, a configuration with fixed and moving flanges, shallow, wide groove and clamps, similar to the one described
for the blade model is used.

Since only two "blades" are required for the experimental facility, the supporting structure resembles a flat paddle wheel (Fig. 2, Fig.17) more than a turbine disc. Made of 12.7 mm (0.5 in.) thick 17-4PH stainless steel plates, it supports the blade model and calibrating body at the proper radius. Tapped holes and threaded studs are used for mass addition, for two plane balancing.

By design, one of the plates can be easily removed from the shaft (Fig. 3) to allow access to shaft mounted instrumentation. To avoid consequent loss of alignment or balance, shear loaded pull-out dowel pins are used instead of fasteners in critical locations.

The facility was initially conceived to operate at 800°K and, in order to avoid carrying hot gases through the shaft and seals, a counterflow heat exchanger is mounted on the shaft. The final design choice was dictated by the following requirements:

i) Small overall size with large length-to-diameter ratio.

ii) Small pressure drop on exhaust gas side.

iii) Ability to withstand high centrifugal loads.

iv) Feasibility of manufacturing and assembly.

v) Ease of partial disassembly to permit inspection and/or repairs of the instrumentation.

The especially designed heat exchanger consists of four identical modules connected in series (Fig. 2, 11) and arranged in two pairs on both sides of the shaft. The
incoming cooling gas is flowing inside 61 small diameter (1.8 mm (0.072 in.) OD/1.4 mm (0.054 in.) ID) stainless steel tubes. The outgoing exhaust gas is flowing in the space between them and a large diameter (38.1 mm (1.500 in.) OD / 28.6 mm (1.125 in.) ID) stainless steel tube, which is the shell of the heat exchanger. Heat is exchanged between the two streams and, as a result, appropriately preheated gas reaches the blade model, whereas the temperature of the exhaust gas at the seals is well below their working limit. The components of one such module are shown in Fig.12.

The overall length-to-diameter ratio is 1780 and the calculated effectiveness at design point is 0.88. Four O-Ring seals permit removal of two heat exchangers and one supporting structure plate for access to shaft mounted instrumentation.

3. 4 Current Supply System.

The uniformly distributed heat sources of Sect.3. 1 were implemented by resistive heating. A 400 Ampere welding rectifier (WESTINGHOUSE model WSH LG) supplies DC current which is conducted onto the rotating shaft by means of especially designed tubular conductors, vacuum feedthroughs, carbon brushes and copper power slip rings (Fig. 2, Fig.17.)

The trailing type carbon brushes (25.4x 38.1 mm (1x1.5 in.) cross section) are made for operation in vacuum (STOCKPOLE grade 566.) The copper power slip rings (38.1 mm (1.5 in.) wide, 127 mm (5.0 in.) OD) are designed for minimum axial length, minimum diameter and key-less grip on the shaft.
Two flat copper conductors, heat sunk to and supported by the structure plates (Fig. 3,) and sliding electrical contacts, which accommodate geometry changes (Fig. 7, 11) and were mentioned in the previous paragraphs, lead the current to two copper terminals at the hub and tip of the blade model.

The maximum design current is 1000 Amperes, and water cooling for the brushes and conductors is provided for use at this level.

3. 6 Primary Cooling System.

Refrigerant-12 (R-12) was chosen to be the cooling gas, according to the reasoning of Sect.2. 3. A clean, oil - free thermodynamic - mechanical compression system, designed for a maximum flow rate of 0.05 kg/sec (0.1 lbm/sec,) is used to recirculate R-12 in a closed loop (Fig.20.)

The hot R-12, after passing through precooler, is liquefied in the condenser at 236 oK (-350F), subcooled to 198 oK (-193 oF) in a dry-ice / R-11 bath, and pressurized as a liquid by a small gear pump (MICROPUMP model 120-401-10A.) Still liquid it passes through the flowmeter and then is expanded through a metering valve and evaporated in two water heated exchangers in order to be recycled through the shaft mounted heat exchangers and blade model.

3. 7 Secondary Refrigeration System.

The condenser of the primary R-12 system (Fig.20) is an especially designed tube and shell counterflow heat exchanger,
made out of standard copper tubing and fittings. It constitutes also the evaporator of a low temperature (236 oK (-35 oF)) refrigeration system. A 3.7 kW (5 hp) water cooled, oil lubricated commercial compressor (DUNHAM-BUSH model WH51PL) pumps Refrigerant-502 (R-502) through the secondary loop.
CHAPTER 4

INSTRUMENTATION

4. 1 Infrared Radiometry.

The temperature distribution on the surfaces of the blade model and calibrating body is measured by means of an infrared radiometer, designed, optical bench aligned and tested by the author at M.I.T. (Fig. 5.) Positioned inside the vacuum chamber with its optical axis on a plane through the shaft centerline, it "sees" alternatively the blade model and calibrating body pass by as the shaft rotates. Although their velocity is high (on the order of 100 m/sec), the time constant of the radiometer and the associated electronics is high enough (less than 1 μsec) so that it can follow the temperature gradients on the rotating surfaces.

The sensing element of the radiometer is a mercury cadmium telluride (HgCdTe) infrared detector. The element is square (1x1 mm,) mounted on a stainless steel dewar equipped with the standard Irtran-II window (Santa Barbara Research Center, type 40742) and is cooled to 77 oK by liquid nitrogen. It has a spectral response from 5 to 15 μm which peaks to 1.4E+10 cm-1 Hz/Watt for an incident wavelength of 12 μm. Its time constant (not measured) is in the range of 100-800 nsec. Performance curves of such a typical detector are shown in Fig.22, taken from Ref.[33]. Two modifications have been made to this otherwise standard detector:
i) The cold radiation shield of the detector element has been replaced by one reducing the field of view for improved detectivity.

ii) The liquid nitrogen dewar has been modified to allow filling and nitrogen vapour relief with the test chamber under vacuum.

An imaging system is required in order to collect the radiation emitted by the measurement spot on the model surface and focus it on the detector sensing element. The ideal system should:

i) Collect an adequate amount of radiant power to provide a usable signal to noise ratio,

ii) Provide the desired spatial resolution,

iii) Provide a long working distance in order not to interfere with rotating parts,

iv) Introduce the minimal amount of optical aberrations and energy losses and

v) Provide an adequate depth of field, so that no refocussing is necessary for a curved surface scan.

To satisfy as many as possible of these conflicting requirements both refractive and reflective systems have been considered. The final choice of an all - reflective, two - spherical - element, stigmatic optical system, was based on the following reasons:

i) An all - reflective optical system does not suffer from chromatic aberration, since the reflection law is independent of wavelength. This is a substantial advantage, considering
the width of the infrared spectrum in which the detector is sensitive (5 to 15 μm.)

ii) Two reflections, which are necessary for the correction of the spherical aberration, introduce very small losses (less than 5% for the bandwidth of interest). This is so because metals behave as near perfect reflectors in this region of the infrared. Refractive elements on the other hand introduce reflection losses for each interface surface and transmission losses as well, depending on the thickness of the element and the wavelength of the incident radiation.

iii) An all - reflective system can be aligned, tested and focussed using visible radiation.

iv) Reflective elements are readily available at lower cost in large apertures than are infrared refractive ones.

Thus the optical system used is a Cassegrain - type objective, shown in Fig.21. It uses a 200 mm diameter concave spherical mirror as primary and a 90 mm diameter spherical convex mirror as secondary. It is corrected for spherical aberration and, being all reflective, is free of chromatic aberration. All other aberrations are negligible due to the fact that the object (i.e. measurement spot) is on - axis and is essentially a "point source" (a square of 1.6 mm side).

The whole imaging system could be scaled up or down geometrically without any effect on effective numerical aperture or angular magnification. The choice of dimensionally large optical components was dictated by the requirement of long working distance and by the tolerance in
alignment and focussing they provide.

The requirement of adequate depth of field was sacrificed, despite the convenience associated with it, in favor of a large aperture which is essential for a large signal to noise ratio. Instead, the radiometer is focussed on each measurement spot.

The low level detector signal passes through an ultra low noise video preamplifier (PERRY AMPLIFIER model 666, 40 dB gain, noise figure 2dB, bandwidth 20Hz – 3MHz) located inside the vacuum chamber and is then transmitted through a coaxial cable to the digital oscilloscope (see Sect. 5. 2)

The modulation or "chopping" of the detector incident radiation, as well as scanning in the angular direction are accomplished by the rotation of the blade model and calibrating body. Scanning in the radial direction is achieved by means of a precision linear translation stage (R-stage). A similar stage is used for positioning of the radiometer assembly as a whole in the axial direction, providing thus a means of focussing (Z-stage). This compensates for the curvature of the leading edge and the limited depth of field of the imaging system and minimizes the measurement spot size on the surface for better spatial resolution of the temperature field. A brief description of the stages is given in Sect. 5. 2.

4. 2 Contact Temperature Measurements.

Temperature is measured by thermocouples in eight
locations inside the blade model and calibrating body:

i) blade model skin at the tip,
ii) " " " " hub,
iii) calibrating body at the tip,
iv) " " " hub,
v) supply plenum at the tip,
vi) " " " hub,
vii) impingement space at the tip,
viii) " " " hub.

Some of the blade model thermocouples, are shown in Fig.18. Standard calibration Chromel - Constantan (type E) 0.1 mm (0.004 in.) diameter wires in a 0.5 mm (0.020 in.) diameter stainless steel sheath and packed MgO insulation are used. All sheaths are guided through the instrumentation spine of the blade model (Fig.18) or appropriately located grooves and exit to the vacuum together with the pressure tubes through a sealing plug in the center of the moving blade model flange. Locating pins and an O-ring seal allow for a slight radial motion of the sealing plug while inhibiting relative rotation with respect to the model.

For support against centrifugal loads and vibration, all thermocouple sheaths and pressure tubes are brazed on the instrumentation strut (Fig. 3, 4.) This is a flat strip of stainless steel 1.0 mm (0.040 in.) thick and 9.5 mm (0.375 in.) wide, extending from the blade model base to the shaft surface. It is cantilevered on the instrumentation block at the shaft end and the above mentioned sliding support at the
other end accommodates thermal expansion. This configuration allows change of the blade model stagger angle by a simple rotation of the moving flange and a consequent twist of the instrumentation strut with no other changes whatsoever. The same principle is used for the two calibrating body thermocouples.

To minimize spurious electromotive forces introduced by alloy transitions at different temperatures in the thermocouple loops, while keeping the number of slip rings dedicated to temperature measurement as small as possible the "cold junction" of all thermocouples is located on the shaft. A solid state, precision temperature transducer (AD 590M) measures the temperature of an aluminum isothermal instrumentation block (Fig. 4.) Solder pads, bonded on this block, provide a convenient location for splicing thermocouple with copper wires at a measured cold junction temperature. This temperature is used for all thermocouple output compensation, both on line, for temperature monitoring through the computer video terminal and later on, during the data reduction process.

All thermocouples, except the one at the blade model skin tip share a common ground. Thus 12 of the total 16 slip ring channels accommodate eight thermocouples, and one solid state temperature transducer. The particular thermocouple is left floating in order that the potential difference between its (-) wire and the (-) wire of the thermocouple at the skin hub would provide a measure of the current through the
resistive skin. Unfortunately, both model skin thermocouples were damaged during final assembly. The sheath and (+) wire of the tip thermocouple failed, making temperature measurement impossible and reducing the confidence in the current measurement method described above. Both thermocouples had to be spliced, because of a model repair. Since the splices are at unknown and not necessarily equal temperatures, an appreciable error can be introduced, which is believed to be the source of inconsistency between the skin thermocouple at the hub and the radiometer temperature readings.

4. 3 Pressure Measurements.

Pressure measurements have been achieved in five locations inside the cooling passages of the rotating model (Fig.18) providing a first insight about supply, discharge, stratification, and stability effects. Pressure is measured at:

i) supply plenum at hub,

ii) exhaust " " hub,

iii) supply " " tip,

iv) impingement space at hub,

v) " " " " .

The scarcity of available slip ring channels, the steady state nature of the experiment and the requirement of interchangeable impingement inserts, made the use of a single transducer in a shaft mounted pressure scanning valve particularly attractive.
Each pressure tap inside the model is flush with the wall surface, measuring thus static pressure which is transmitted to the pressure scanning valve by means of a 5.8 mm OD / 3.4 mm ID (0.032 in. ID / 0.016 in. OD) stainless steel tube. All five tubes are brazed on the instrumentation strut (see Sect.4. 2) for mechanical support and temperature equalization, so that differential pressure measurements between ports introduce minimal error. Absolute pressure measurements are compensated for the gas column effect, using as inputs measured temperatures on the shaft, impingement plenum hub and tip (App. 2). A tube splice close to the shaft, allows blade model replacement without any pressure scanning valve disassembly or removal.

The especially designed pressure scanning valve, shown in Fig. 9 and 23, consists of:

i) A single pressure transducer (KULITE model CQ-080-50, 0 - 50 psig range) mounted with its diaphragm normal to the axis of rotation and at a radius of 52.4 mm (2.063 in.,)

  ii) A hollow stainless steel plunger which moves linearly and scans all pressure ports,

  iii) A sheet metal cover which supports the temperature compensating module of the pressure transducer and the flexible connecting wires against centrifugal forces, and

  iv) The main valve body.

The plunger is manually actuated from the outside of the vacuum chamber by means of a double acting master - slave hydraulic cylinder system, shown in Fig.10. Vacuum pump oil
is used as the system fluid so that minor leaks do not contaminate or deteriorate the vacuum of the chamber. A ball bearing, acting in a way similar to the throwout bearing of an automotive clutch, allows remotely controlled linear motion of the plunger even when the shaft is rotating. A linear potentiometer, with the windings fixed on the chamber frame and the wiper attached on the slave cylinders, provides information about the exact location of the scanning port.

4.4 Calibrations.

i) Infrared radiometer calibration: The radiometer was checked for short and long term drift and repeatability. Positioned at the radius corresponding to the tip calibrating body thermocouple, it scanned continuously the same arc on the surface. The power input on the calibrating body heating element was varied and excursions in temperature were made. This process was repeated at different dates and the results are plotted on Fig.24. Each data point corresponds to the mean of four successive measurements, taken during four successive revolutions of the shaft. The repeatability of the radiometer is better than this plot shows. The small discrepancies observed at the higher temperatures are due to thermal capacitance between the measuring thermocouple and the foil surface, which introduces a time lag in the readings of the radiometer. This claim is substantiated by examining a number of four successive radiometer measurements: the typical standard deviation, shown on Fig.25, is 0.004 volts.
for a full scale setting of ±1 volt, which corresponds exactly to the resolution of the digital oscilloscope (see Sect. 5.2.)

For each geometrical configuration of the blade model, a series of preliminary calibration experiments must be conducted. For these, the electrical power lead to the blade model is disconnected, the calibrating body is stabilized at a temperature and a full scan of its surface is conducted. This is repeated for a range of calibrating body temperatures, spanning the expected temperatures on the model skin. The calibrating body is quasi-isothermal. Temperature differences of 2 oK to 15 oK (3 oF to 27 oF) depending on the electrical power input to its heating element, have been measured across its ends, due to conduction to the supporting structure. Its temperature distribution is computed using the one dimensional approximation of a thermally conducting cylinder with a linear heat source along its axis and the known temperatures at two known axial locations. Thus, for each particular spot on the calibrating body—and its corresponding point on the blade model—surface, a set of data pairs \([V, T]\), each corresponding to a calibration scan at a different calibrating body temperature is generated (Fig. 26,) where:

- \(V\) is the signal of the radiometer at the spot.
- \(T\) is the corresponding spot temperature, computed using as inputs to the above one dimensional scheme the measured end temperatures of the calibrating body, its heating
current input and the known spot coordinates.

This set of data points constitutes the calibration data set for the particular spot. Such a set is generated for all measurement spots and used during the data reduction process, as will be described in Sect. 6.3.

ii) Temperature transducer and thermocouple calibration: The reference temperature transducer (see Sect. 4.2) was calibrated against a precision mercury thermometer (0.1 oK resolution) at ambient temperature. The output of all thermocouples (type E, standard calibration) was verified to be within 0.1 oK of the mercury reading at ambient.

iii) Pressure transducer calibration and compensation: After the final mechanical and electrical assembly, the pressure transducer was calibrated under static pressure and no rotation against a precision Bourdon type pressure gage (ACCO Helicoid Gauge model 2514-Ø.)

Any DC drift and centrifugal effect on the pressure transducer are compensated by scanning the last port of the pressure scanning valve. Being open to the chamber vacuum, it provides a baseline measurement used later for the pressure data reduction to offset all other pressure measurements. Raw pressure measurements are also corrected for rotational effects according to the procedure described in App. 2.
CHAPTER 5

DATA ACQUISITION

5.1 Introduction.

Data acquisition is controlled and accomplished by a minicomputer, requiring a minimal amount of human intervention. This was deemed necessary because of:

i) The large amount of data acquired during a single experiment,

ii) The precise timing required for accurate spatial resolution of the radiometer measurements,

iii) The multitude of parameters which must be monitored during the experiment set-up and run.

5.2 Computer Controlled Data Acquisition.

The components of the computer controlled data acquisition system, shown schematically in Fig.27 are:

1) A Digital Equipment Corporation MINC-11 minicomputer (updated later to a faster MINC-23,) which is used as the main controller, monitor, data display and storage device.

2) An Analog Devices μMAC 4800-333 12 channel analog to digital converter and multiplexer, which is used for the measurement of:

i) Pressure scanning valve position (linear potentiometer voltage,)

ii) Temperatures inside the model and calibrating body
(thermocouple voltages,)
i) Reference temperature of thermocouple "cold" junctions
(one solid state temperature transducer output,)
iv) Current through blade model skin (voltage drop across
the span,)
v) Pressure in the cooling passages of the model and in the
vacuum chamber (pressure transducer output,)
vi) Current through blade model and calibrating body
(voltage drop across the power supply shunt.)

The device communicates with the computer through an
RS232C serial interface at 9600 baud.

3) A 16 channel instrumentation slip ring set (MTC model
BH-16) which transmits signals from the rotating transducers
(thermocouples, AD590M, pressure transducer) to the above
analog to digital converter - multiplexer.

4) A Nicolet EXPLORER III digital oscilloscope with the
model 204-A plug-in unit (2 channels/20 MHz time base) and the
NIC-2081 IEEE-488 (GPIB) interface accessory. The function of
each channel is described below:

Channel A: Receives the preamplified, analog detector
signal (see Sect.4.1) during four successive revolutions of
the shaft. Filters out frequencies above 1MHz and samples it
at the rising edge of the timing pulses coming from the
encoder - timing / counting circuit, whose description
follows. Converts it to digital with an 8 bit or 0.4%
resolution and stores it in a 4096 word random access memory
and finally, displays the signal on the oscilloscope screen.
Channel B: Receives the external trigger and timing / sampling pulses. This signal is not filtered, stored nor displayed.

Upon command, four triads of data points, corresponding to the four successive revolutions of the shaft, are transferred to the computer memory. In each triad:

1) The first data point, at shaft phase angle \( \theta \), corresponds to the radiometer in-focus spot on the model surface.

2) The second, at shaft phase angle \( \theta + 90^\circ \) degrees), corresponds to background or reference level.

3) The third data point, at shaft phase angle \( \theta + 180^\circ \) degrees), corresponds to the radiometer in-focus spot on the calibrating body surface.

5) An optical rotational encoder (BEI Electronics model L25G-500-ABZ-7404-ED15-S) mounted at the far end of the shaft (Fig.10 and Fig.17.) Two channels in quadrature (A and B) with 500 pulses per revolution plus a third reference channel (Z) with 1 pulse per revolution provide TTL pulses synchronous to the shaft rotation.

6) An timing / counting digital circuit, especially designed and manufactured by the author, combines pulses from channels A and B of the rotational encoder to produce TTL pulses synchronous to the shaft rotation, with an angular resolution of \( 1/2000^\circ \)th of a revolution or \( 0.18^\circ \) degrees. Thus, the rotational encoder - timing / counting circuit system locks the oscilloscope sampling to the shaft angle and
maintains synchronization regardless of the level or fluctuations of angular velocity.

Using the reference pulse of channel 2, synchronous digital counters and a gate, it allows only the first 1024 encoder pulses of each revolution to reach the trigger/timing input of the digital oscilloscope. Thus, background data corresponding to the remaining 976 pulses of each revolution are not stored, and the 4096 word random access memory of the oscilloscope is filled after exactly four successive revolutions.

Finally, this circuit provides a flag for the computer, so that the oscilloscope is never instructed to start sampling in midst of the 1024 timing pulse train.

For successful radiometer measurements, timing is of paramount importance. Since data points are stored sequentially in the oscilloscope random access memory and their ordinal number corresponds to the angular coordinate theta, any spurious or missed timing pulses result in a loss of synchronization and invalid measurements. The timing circuit, operating in the proximity of two welding rectifiers and a number of AC and DC motors has performed very well, limiting the fraction of rejectable measurements per test to less than 1% on the average. These measurements are smoothed out during the data reduction process (see Sect.6.2.)

7) Two precision, motor driven, linear translation stages, shown in Fig. 5, used to position the radiometer in the radial (R) and axial (Z) directions. The R (Z) stage has a step
resolution of 0.0127 mm (0.0005 in.) [0.0254 mm (0.001 in.)] and a total travel of 203.2 mm (8.0 in.) [63.5 mm (2.5 in.) respectively.] Spring loaded nuts on the lead screws and preloaded radial and linear ball bearings on hardened rod slides minimize axial backlash or side play.

Stepping motors with 200 steps/revolution (SLO-SYN model HS25) are used as drivers. During the experiment they are stepped under computer control at maximum torque speed (250 steps/sec) through a 16 bit parallel digital input / output interface board (Data Translation DT 2768) and two translator boards (SLO-SYN model STM 101) which convert computer pulses to motor steps. Three bits lock / unlock, forward / reverse and step each motor.

Panel mounted controls and display counters allow stage positioning, single stepping, speed and acceleration variation under manual control.

5. 3 Manual Data Acquisition.

Four quantities are measured and manually recorded several times during each experiment. These are:

1) Shaft rotational speed: A digital frequency counter (HP model 5314A) is connected to channel A of the optical rotational encoder output, and displays the shaft rotational frequency multiplied by 500 in kHz. The measurement mean is used for the computation of test Rossby number and the compensation of raw pressure data for centrifugal effects (see App. 2.)
2) Vacuum chamber pressure: Is measured through an electronic vacuum gauge (MAGNEVAC model GMA 201.) The measurement taken when the last port of the pressure scanning valve (which is open to vacuum) is scanned, is used to compensate raw pressure data for zero offset (see App. 2.)

3) Coolant volume flow: Is measured through a flowmeter (FISHER and PORTER model 18A3555A.) The measurement mean is reduced to mass flow after density corrections, according to the measured flowmeter temperature (see below) and its manufacturer's recommendations.

4) Flowmeter temperature: Is measured by means of an Iron - Constantan (type J) thermocouple through a digital temperature readout (OMEGA model 199-JF-X-X-DSS.) with a $0.5^\circ$C ($1^\circ$F) resolution.
CHAPTER 6

DATA REDUCTION

6.1 Introduction.
During the experiment raw data is stored on an 8 in. floppy disc for later processing and reduction. After the first processing step, which consists of signal conversion to MKSA units and calculation of statistical quantities, data is transferred to the Digital Equipment Corporation PDP 11/70 computer of the Gas Turbine Laboratory. There, taking advantage of the large storage capacity, available software and high speed of processor and peripherals, data is reduced through the following steps.

6.2 Data Smoothing.
As mentioned in Sect.5.2, occasionally synchronization during the experiment is lost and an invalid radiometer measurement results. Since this occurrence is rare (less than 1% on the average,) the invalid measurements are replaced by the average value of all neighboring points.

6.3 Radiometer Voltage to Temperature Conversion.
After any scan, the data consists of radiometer output voltages. The calibration data set of each spot (see Sect.4.4) is used for linear interpolation and conversion of voltage to temperature (Fig.26.) In this way effects due to
geometry, directional and spectral emissivity, spectral responsivity and nonlinearities of the detector, and imperfections of the imaging system are factored out.

6.4 Computation of the Local \( h \) and \( \text{Nu} \).

The local heat transfer coefficient \( (h) \) is computed by means of a finite element power balance scheme, described in detail in Sect.3.1 and illustrated by Fig.16, using as data the heat input per unit area to the blade model resistive skin, the local spot temperature \( T_0 \), the temperature of all its neighboring spots \( T_i \) (\( i=1,4 \)), the cooling gas supply temperature \( T_c \) and the radiation loss to the surroundings:

\[
h = \left( \frac{2}{\text{Af}} + \frac{k}{\text{Sl}} \sum_{i=1}^{4} \frac{S_i}{\text{Si}} (T_i - T_0) \right) \frac{\epsilon \mu}{\text{To} - \text{Tc}} \text{ Di} \quad \text{(Equ.6.1)}
\]

The local Nusselt number \( (\text{Nu}) \) is then computed by use of the equation:

\[
\text{Nu} = \frac{(h \, d)}{k} \quad \text{(Equ.6.2)}
\]

where: \( h \) = local heat transfer coefficient \( \text{(W/sq.m oK)} \)

\( d \) = jet diameter \( \text{(m)} \)

\( k \) = thermal conductivity of Refrigerant-12 at coolant supply temperature \( \text{(W/m oK)} \)

The data reduction sequence described in Sect.6.3 and 6.4 is illustrated by Fig.28.
6. 5 Averages and Non - Dimensional Parameters.

Spatial and temporal averages are computed for data presentation and comparisons with other researcher's measurements. The conventions used for data presentation can be found in App. 1.

Contact temperature time averages are computed and normalized by the coolant supply temperature.

The corrected, time averaged pressure is computed for each pressure scanning valve port and normalized by the coolant supply pressure.

The blade model surface is divided, for comparison purposes, into seven (7) imaginary, equal width, radial strips (see Fig. 29.) The first strip starts at $X = \theta.0$ and the last ends at $X = X_{max}$ on the unfolded skin. Each strip is divided in two sections: one corresponding to the part of span cooled by the impinging jets, and another, corresponding to the rest of the span which is cooled by convection. Thus, strips #1 to #7 correspond to the full skin span, while strips #8 to #14 to the impingement cooled span. The "stagnation" strips correspond to #4 and #11.

Average temperatures are computed for the impingement cooled span and the whole blade model surface and normalized by the coolant supply temperature.

Nusselt numbers are computed for each part - span and full - span strip and the whole blade model surface. A \[1/\cos(a) \times S1 \times S2\] weighting is used, where (a) is the slope of the folded skin at the measurement spot, and S1 \times S2 the spot
surface area for average temperature and Nusselt number computations.

Finally, non-dimensional parameters, such as Reynolds, Rossby and Mach numbers, based on corrected mass flow, average rotational speed and referenced to the coolant properties at the average supply temperature and pressure, are computed for both plenum inlet and impingement jets.
CHAPTER 7

EXPERIMENTAL PROCEDURE AND RESULTS

7.1 Experimental Procedure.

Each test consists of four tasks:

Task 0: Monitor. This is the default task. The data from the analog to digital converter - multiplexer is displayed on the video terminal screen. No data is stored. Used during experiment set-up and stabilization, it can be interrupted when any of the following three tasks is selected by the user.

Task 1: Calibration. The radiometer is positioned at the calibrating body tip thermocouple radius and scans discrete spots on an arc on the blade model / calibrating body surface, by motion of the Z-stage. This sequence is repeated continuously until the user terminates it through the video terminal keyboard. Data aquired through this task is used for radiometer drift check and compensation and rotational speed / radiometer output correlation, neither of which was observed (Sect.4.4) during the tests.

Task 2: Test. The radiometer scans discrete spots on the blade model and calibrating body surface, starting from the upper right corner of the measurement window and using the fastest possible motion sequence of the R and Z stages. This task requires approximately 10 min. to be completed.

Task 3: Recalibration. Identical to calibration,
performed after the scan for the same reasons.

Before each blade model scan the desired mass flow is established through the cooling system and the heating current input is progressively increased to its final value. The facility is allowed enough time (approximately 1 hr.) to reach thermal equilibrium. The thermocouple temperatures are monitored through the computer video terminal under Task \# 6 and thermal equilibrium is considered to be reached when the temperature change rate is less than 1 oK (2 oF) per 10 mins.

The driving motor is then switched on, and the shaft is allowed enough time (3 to 5 mins) to reach the constant test speed. Tasks \# 1, 2 and 3 are then performed in succession, in approximately 15 mins. During this time the two operators perform all manual data acquisition tasks, discussed in Sect.5. 3, operate the pressure scanning valve through the hydraulic actuator system (see Sect.5. 2) and monitor the condition of critical facility components.

Data is stored on an 8 in. floppy disc. Two data files are created during each of the tasks 1, 2, 3. One, formatted, provides a soft disc label, test parameter data and oscilloscope settings. The other, unformatted, contains transducer and radiometer data.

7. 2 Experimental Data.

Twenty-four (24) measurements, originating from various transducers are taken during the test for each spot scanned. After the data reduction process, the number of measured and
computed variables rises to forty-three (43) per spot (see App. 1.) Since there are 440 such spots for the geometry tested, the total number of measurements is 18,920 per test.

To facilitate data analysis and presentation, a large amount of software was developed. It allows X-Y plotting of any variable against any other of the 43 and superposition of plots originating from the same or different tests. Value and contour plots can be readily produced for both folded and unfolded blade model skin, to provide a visual representation of heat transfer phenomena occurring under rotation.

Parametric study: A series of eighteen (18) experiments was conducted in order to investigate the effects of rotation on impingement cooling. A single blade geometry, described in detail in Sect. 3. 3 and shown in Fig. 18 and 19, was used for all tests listed here. The nominal values of the physical quantities varied were:

i) Coolant (R-12) mass flow (0.005, 0.010 and 0.020 kg/sec.)

ii) Rotational speed (850, 1450 and 2000 RPM.)

iii) Current input to the blade model skin (31.6 to 60.6 Amperes.)

| TABLE # 2 |
| TEST PARAMETERS |
| Re jet (average) | 17000 | 36000 | 74000 |
| Rotational speed (% of Nd) | 43 | 75 | 100 |
| Temperature ratio Tw/Tc | 1.28 to 1.38 |

The resulting matrix of nominal, non-dimensional
parameters is listed in Table & 2. The average jet Reynolds number \( \text{Re} \) is by definition:

\[
\text{Re} = \frac{4 \dot{M}}{N \mu \pi d}
\]  

(Equ.7.1)

where: \( \dot{M} \) = flowmeter mass flow \( (\text{kg/sec}) \)
\( d \) = jet hole diameter \( (\text{m}) \)
\( \mu \) = R-12 viscosity at Tc \( (\text{N sec/sqm}) \)
\( N \) = number of jets

The facility "design" rotational speed \( \text{Nd} \) of 2000 RPM simulates, according to Equ.2.26, a speed of 30000 RPM of the full scale turbine blade. The temperature ratio is computed using the impingement cooled surface area weighted average as \( T_w \) and the time average value of coolant supply temperature as \( T_c \). Since this ratio could be determined only after the experiment, constant values of heating current were used, which resulted in the above range of \( T_w/T_c \).

It should be noted at this point that these values of \( T_w/T_c \) cover the lower operating range of real gas turbines (1.5 to 2.0). Further increase of the heating current input would expose the current version of the blade model to the risk of epoxy overheating at the upper part of the skin which is not impingement cooled, while further reduction of the coolant temperature drives the coolant supply system unstable.

The experimental data is presented in Fig.33 to 44. Table # 3 provides a quick reference for the nominal conditions of each test.
TABLE # 3
TEST CONDITIONS

<table>
<thead>
<tr>
<th>% of Nd (RPM)</th>
<th>Test 043</th>
<th>Test 057</th>
<th>Test 063</th>
</tr>
</thead>
<tbody>
<tr>
<td>43 (850)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>74 (1450)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>100 (2000)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Low Re, low Tw/Tc</th>
<th>Test 042</th>
<th>Test 056</th>
<th>Test 062</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Low Re, high Tw/Tc</th>
<th>Test 039</th>
<th>Test 055</th>
<th>Test 065</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Med. Re, low Tw/Tc</th>
<th>Test 044</th>
<th>Test 054</th>
<th>Test 064</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Med. Re, high Tw/Tc</th>
<th>Test 040</th>
<th>Test 059</th>
<th>Test 066</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>High Re, low Tw/Tc</th>
<th>Test 041</th>
<th>Test 058</th>
<th>Test 067</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>High Re, high Tw/Tc</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
</table>

The symbols and abbreviations used for data presentation are explained in App. 1. One page summaries of geometrical constants, time and space averages and non-dimensional quantities are presented for each test in App. 3, arranged according to test serial number to facilitate retrieval by the reader. The data is presented in the form of temperature (Fig. 33 - 38) and Nusselt number (Fig. 39 - 44) contour plots in groups of three tests, corresponding to the same nominal Re and electrical heating power input, and increasing rotational speed. On these plots, the outline of the unfolded blade skin is denoted by the rectangle. The tip of the model corresponds to the top and the sense of rotation is from left to right. The jet centerline traces correspond to the cross intersections. The sample window, limited by the line of sight of the radiometer and the physical boundaries of the skin, is shown as a polygon.
Test # 835 was conducted with the blade model heated and the cooling loop evacuated, to provide an estimate of the conduction losses and to observe any substantial irregularities in thickness or emissivity of the model skin. The hot spots observed on the temperature contour plot (Fig.30) are caused by a reversed flow of R-11, originating from shaft seal leakage, through the jet holes. Tests # 860 and # 861 are repetition runs of tests # 855 and # 856 respectively, conducted to check the rig resettability. Nusselt number distributions of the surface and along a radial line corresponding to theta=7 are presented in Fig.31 and 32 respectively. Test # 837 is the first successfully completed scan of the blade model skin with cooling, and is included here for comparisons.
CHAPTER 8

ERROR AND UNCERTAINTY ANALYSIS.

8.1 Temperature Measurement Uncertainty.

The solid state temperature transducer (AD599M,) used to measure the reference temperature of all thermocouples, was system calibrated according to the procedure described in Sect.4.4. Since the shaft temperature is within 1.0 K of the calibration temperature, a potential source of absolute error (1.0 K) is eliminated. The combined effect of nonlinearity (0.3 K) and repeatability (0.1 K) result in an uncertainty of 0.3 K in reference temperature.

All the thermocouples used are Chromel - Constantan (type E, standard calibration) with an error limit of 1.7 K. The overall error, including the 0.3 K uncertainty in reference temperature, is 1.7 K.

The radiometer is calibrated by means of the calibrating body thermocouples, which introduce an uncertainty of 1.7 K. The oscilloscope contributes a digital conversion error of 0.5 K and an estimated 1.0 K is attributed to emissivity variations, deviations of the temperature distribution on the calibrating body from the one computed, etc. The overall uncertainty of radiometric temperature measurements is estimated to be less than 2.5 K.
8.2 Pressure Measurement Uncertainty.

Two of its most important sources are eliminated through the experimental technique used. First, by scanning the sixth (open to vacuum) port, zero balance uncertainty (3% full scale) and zero shift due to rotational g - field are accounted for in the pressure correction process (see App. 2.). Second, by allowing approximately 1 hr. for thermal equilibrium, so that temperature changes are less than 1 °C (1.8 °F) per 16 min., transducer sensitivity (1.5%/100 °F) and no load output (0.5%/100 °F) variations with temperature are minimal. When these and all other sources of uncertainty, such as combined non - linearity and hysteresis (0.5% full scale,) repeatability (0.15%), transverse acceleration sensitivity (0.00003% full scale/g,) amplifier gain (0.65%), analog to digital converter resolution (13 bit or 0.01%) and excitation voltage variation (0.5 V/12 V or 4%) are combined, the probable error in raw pressure measurement is:

<table>
<thead>
<tr>
<th>Measurement: N/sq m (% FS)</th>
<th>Uncertainty: N/sq m (% FS)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.86E+05</td>
<td>0.038E+05</td>
</tr>
<tr>
<td>(25)</td>
<td>(1.1)</td>
</tr>
<tr>
<td>1.73E+05</td>
<td>0.072E+05</td>
</tr>
<tr>
<td>(50)</td>
<td>(2.1)</td>
</tr>
<tr>
<td>3.45E+05</td>
<td>0.138E+05</td>
</tr>
<tr>
<td>(100)</td>
<td>(4.1)</td>
</tr>
</tbody>
</table>

The centrifugal column effect is less than 10% of the corrected pressure measurement, and therefore its contribution to the overall probable error through small temperature uncertainties is negligible.
The accuracy of pressure measurements can be improved by modification of the transducer excitation source, since the 12 V gel-cell battery currently used is the largest source of uncertainty.

8. 3 Mass Flow Uncertainty.

The flowmeter used for all tests (Sect.5. 3) is accurate within 2% of full scale. The mass flow stability for all tests is typically within 0.5% of the average measured value with a standard deviation less than 1%. The flowmeter temperature stability is better than 0.5 oK (1 oF), equal to the resolution of the thermometer used (see Sect.5. 3,) which results in a density variation of 0.2%. The combined effect on the average jet Reynolds number is an uncertainty of 22%, 10% and 5% for Re equal to 17000, 36000 and 74000 respectively.

The accuracy of the mass flow measurements can be improved by use of an array of flowmeters, each to be used close to its maximum capacity. The mass flow uncertainty in the data listed here can be reduced by post-calibration of the flowmeter used against such an array.

8. 4 Rotational Speed Stability.

Although there is some deviation of each test speed from the corresponding nominal, rotational speed is steady, typically within 0.4% of the average measured.
8. 5 Heat Transfer Coefficient Uncertainty.

In order to understand the sources of uncertainty in the heat transfer coefficient, it is necessary to examine each term of Equ.6.1.

The radiation loss term is very small, less than 1% of the electrical power input, and its contribution to the overall error is negligible.

The conduction loss term is proportional to the net algebraic sum of all temperature differences. If the ratio $Q_{cond}/Q_{select}$ is evaluated for various input current values used and a 1 oK net sum of temperature differences, Table & 5 results.

<table>
<thead>
<tr>
<th>TABLE &amp; 5</th>
<th>CONDUCTION LOSS CONTRIBUTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model Current (Amp)</td>
<td>3Ø</td>
</tr>
<tr>
<td>$Q_{cond}/Q_{select}$ (per oK)</td>
<td>0.033</td>
</tr>
</tbody>
</table>

Therefore, the contribution of the conduction loss term to $h$ (and thus, to the uncertainty of $h$) is variable, and becomes significant for low Re, low power input tests. It is most significant for spots corresponding to local temperature minima (centers of jet traces) or maxima (spots between two jets), where the temperature difference sum takes large values, on the order of 2Ø oK.

It should be noted here that, in order to evaluate this term, data differentiation is necessary. Since however the points are adjacent and the same method (radiometry) is used for their temperature measurement, any systematic errors
cancel out.

The most significant is the power dissipation term. Since the quantities required for its evaluation are current, skin resistance and area, which are measured accurately, its contribution to the overall error is small.

Another source of uncertainty is the denominator (To-Tc). Since To is measured by the radiometer and Tc by a thermocouple, systematic errors do not necessarily cancel out and the difference is accurate within 3.0 oK. For the typical point on the measurement surface (To-Tc)=150 oK, and thus this contribution to the overall uncertainty is small. For the jet traces however, where (To-Tc)=30 oK, the contribution is much larger.

Fig 45 illustrates and quantifies the above arguments. The spot Nusselt numbers along a radial line corresponding to theta=7 are plotted against radius or record number, for three tests at different Reynolds numbers. Curve #1 corresponds to the results of a simple one-dimensional scheme, which neglects conduction and radiation terms in Equ.6.1. Curve #2 corresponds to the results of the full three-dimensional scheme of Equ.6.1 and is the one used for all heat transfer results presented here. The difference between the two is equal to the magnitude of the terms neglected by the first scheme.

Finally, to quantify the typical and maximum probable errors, Equ.6.1 is differentiated and the result is evaluated for the uncertainties of Table # 6 and the resulting
uncertainty in h is listed in Table 6.

<table>
<thead>
<tr>
<th>Variable or constant</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>(30 - 60 Amperes)</td>
</tr>
<tr>
<td>Rs</td>
<td>(59 mΩ)</td>
</tr>
<tr>
<td>k</td>
<td>(16.74 W/m oK)</td>
</tr>
<tr>
<td>t</td>
<td>(97 um)</td>
</tr>
<tr>
<td>Tc</td>
<td>(317 - 342 oK)</td>
</tr>
<tr>
<td>To</td>
<td>(335 - 550 oK)</td>
</tr>
<tr>
<td>Σ (Ti - To) i=1,4 (6 - 20 oK)</td>
<td>1.0 oK</td>
</tr>
</tbody>
</table>

Thus for the typical point on the blade model surface, h uncertainty is less than 3%. The maximum uncertainty is 12% and occurs in temperature field minima, such as the centers of the impingement jet traces.
CHAPTER 9

DATA ANALYSIS AND DISCUSSION

9.1 Comparisons with Other Researchers' Data.

Several investigators have measured spanwise averaged, chordwise resolved heat transfer coefficients associated with impingement cooling. These include Ref.[1] to [7] for leading edge geometries with chordwise exhaust (i.e. with no crossflow) such as the one of Fig.13a, and Ref.[8] to [15] for midchord geometries similar to the one of Fig.13b with the crossflow of the spent air imposed on the jets downstream. In their studies, the above determined how heat transfer varied with coolant flow rate, fluid properties and system geometry and presented their results in the form of graphs or semiempirical correlations of the form:

\[ \text{Nu} = f(\text{Re}, \text{Pr}, \frac{z}{d}, \frac{D}{d}, \frac{p}{d}, \ldots) \]  

(Equ. 8.1)

None of the above accounts for effects due to rotation or concerns a geometry which is exactly the same as the one used for this investigation. Impingement cooling with chordwise exhaust (Fig.13c) imposes a crossflow on the jets, resulting thus in a hybrid of the two other schemes.

In their paper on "Streamwise Flow and Heat Transfer Distributions for Jet Array Impingement with Crossflow", Florchuetz, Truman and Metzger [8] note that:

1) "...for large hole spacings and small channel heights, ...the crossflow provides an increasingly (with mass flow)
significant direct contribution to the heat transfer rate but does not cause a large degradation in the direct contribution from impingement."

ii) "There is some evidence, e.g. flow visualisation, indicating that for the inline arrays the crossflow tends to become channelized between adjacent jet rows."

The geometry tested here can be considered as a single line of a two dimensional array with small jet spacing in the crossflow direction (pitch-x/d = 3) and wide jet spacing in the other direction (pitch-y/d = 16.) Therefore according to the above observations, the effect of crossflow should be minimal. To verify this, the following table is provided for the comparison of the surface average Nusselt number of the tests at lowest shaft speed (43% Nd or 850 RPM) with the correlations of Chupp, Helms McFadden and Brown [1] and Hrycak [2], who conducted experiments for leading edge impingement cooling with no cross flow. It should be noted here that their correlations were extrapolated when necessary to cover the range of parameters tested.

<table>
<thead>
<tr>
<th>Test</th>
<th>37</th>
<th>39</th>
<th>40</th>
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<td>37030</td>
<td>74510</td>
<td>74360</td>
<td>18040</td>
<td>18510</td>
<td>35540</td>
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<td>Chupp, et al.</td>
<td>137</td>
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<tr>
<td>Present work</td>
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<td>332</td>
<td>382</td>
<td>101</td>
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</table>

Low rotational speed measured Nusselt numbers are on the
average 10% above the predictions of Chupp, et al. [1] and 20% below the predictions of Hryck [2].) This agreement with previously published data and the fact that centrifugal effects are proportional to squared, indicate that conducting tests at lower shaft speeds is not necessary.

A number of experimental ([23], [25], [26], [27], [28]) and theoretical ([24], [25], [27]) studies have been published, such as the ones of Morris and Ayhan [23], Mori and Nakayama [24], Mori, Fukada and Nakayama [25], Lokai and Limanski [26], Benton and Boyer [27], and Ito and Nanbu [28] which examine the flow and heat transfer in the simple configuration of a tube or channel, rotating about an axis orthogonal to its axis of symmetry. In all the theoretical studies it has been demonstrated that a secondary flow is created as a result of the Coriolis interaction. Buoyancy forces are not included in the modeling procedure and all predictions suggest improved heat transfer with rotation. Relevant experimental data of Mori, et. al [25] and Lokai et. al [26] support this view. These experiments however have been conducted in regimes where the influence of buoyancy is small. Morris and Ayhan [23] demonstrated that "... for a radially outward flow Coriolis acceleration tends to improve heat transfer. However the Coriolis - induced improvement to heat transfer may be nullified and reversed due to centrifugal buoyancy...". Although their tests were conducted at centripetal buoyancy levels which were higher than any other's reported in the past (Fig.46,) these levels are still well
below the ones encountered in actual gas turbines. The results reported here simulate actual conditions more closely and demonstrate a similar trend (i.e. Nu reduction with increasing buoyancy) for Re = 17000. At higher jet Re however, this trend reaches a minimum and then reverses as buoyancy increases (Fig.47, 48.) A plausible explanation for this will be discussed in the next section.

In the present study, the model surface was divided into seven imaginary, equal width, radial strips (Fig.29,) so that a comparison with the data of Chupp, et al [11], which is spanwise averaged and chorwise resolved by a similar hardware configuration, could be made. The results of such a comparison are plotted in Fig.51. It can be seen that the data demonstrates the same trend with Ref.[11] i.e. a maximum at or next to the stagnation strip which decreases moving towards the trailing edge. The two maxima, substantially off the "stagnation strip", common to the results of Metzger, Baltzer and Jenkins [4] and Dyban and Mazur [5], and attributed to jet instability and "whipping" from side to side in the leading edge cavity by Dyban Mazur and Epik [6], do not appear here. According to the traces, each jet flows towards a preferred direction, defined by the balance of Coriolis, buoyancy and cross flow forces exerted on it.

The radiometric method used, allows finely resolved heat transfer measurements over the blade model skin surface. The measurements of Florochuetz, Berry and Metzger [8], under the abstractions described earlier in this section, provide
"comparable" data resolved in the "spanwise" direction. The periodic oscillation amplitude of Nu, caused by the effect of individual impinging jets, is 90% around its average value for Re=15000 and z/d=2 as shown in Figure 10 of Ref.[8]. The data presented in Fig.52 corresponds to Re=17000 and z/d=2: the peak to peak variation is 80% of the average.

9.2 Discussion.

In order to explain changes induced by rotation, an understanding of the centrifugal, Coriolis and buoyancy effects is necessary. Fig.53 illustrates the accelerations exerted on a fluid particle moving with velocity $V$ with respect to a system of rotating coordinates, and being at a temperature $\Delta T$ above its surrounding particles. The stabilizing and destabilizing effects of Coriolis forces on the boundary layers of the walls of a rotating channel are illustrated by Fig.54, taken from Ref.[30].

1) Effects of rotation on the average Nu.

The surface average Nusselt number, plotted against the mean jet Reynolds number, is shown in Fig.47. A monotonic decrease in Nu with increasing centripetal buoyancy is observed for Re = 17000, demonstrating a similar trend with the experimental data of Moore and Ayhan [23]. At higher Re however, this trend reaches a minimum and then reverses, i.e. Nu increases as centrifugal buoyancy increases. This effect is shown more clearly in Fig.49a,b,c, where Nu is plotted for
all tests corresponding to the same nominal Re. The line and arrows denote increasing centripetal buoyancy.

Similar trends are observed (Fig. 48, 50a,b,c) for the average Nu on the "stagnation" strip, i.e. the strip opposite to the impingement jet holes.

Moore and Ayhan have presented in their paper [23] a plausible explanation for the reduction of heat transfer with rotation for a simple channel flow. For a radial outflow with heat transfer, buoyancy acts on the hot, slowly moving fluid particles of the boundary layer of the wall and tends to slow them down further (Fig. 55a,b,) thus reducing dT/dn and impeding heat transfer. The minimum and subsequent increase in heat transfer observed here (Fig. 47, 48, 49a,b,c, 50a,b,c,) can be explained with a similar argument. At higher rotational speeds (large g fields) and high mass flow (large Re, thin boundary layers \(\Rightarrow\) large \(\Delta T/T\)) buoyancy can cause local flow reversal (Fig. 55c,d) in the boundary layer, which promotes mixing and enhances heat transfer.

ii) Effects of rotation on the local Nu.

The high spatial resolution achieved by the radiometric heat transfer measurement technique used, allows the observation of the behaviour of individual jets and of the effects of rotation on the local Nu distribution.

In the chordwise direction, a monotonic increase [decrease] in temperature [Nu] is observed (Fig. 33 to 38 [Fig. 39 to 44 and Fig. 51]) as the effect of the impinging jets
decays.

In the radial direction, the jet traces are visible on the temperature \([\text{Nu}]\) contour plots of Fig. 33 to 38 [Fig. 39 to 44] as local minima [maxima.] Between jets, the temperature \([\text{Nu}]\) distribution demonstrates local maxima [minima.] It should be noted here that the first jet trace, at the root of the model, does not appear on any of the contour plots, although the pressure measurements, listed on Table \# 9 suggest that it should. The trace of the 12th jet, at the tip, does not appear occasionally for the low and medium mass flow tests, and the pressure measurements of Table \# 9 suggest reversed flow.

At high rotational speed (100% Nd) a steep thermal gradient is present in front of the second jet hole. To appreciate the relative magnitude of these effects, the local Nusselt number is plotted against radius for the radial line corresponding to theta = 7 in Fig. 56 for two tests at the same nominal conditions, but at different rotational speeds corresponding to 43% and 100% Nd. The modulation due to the individual jets is clearly visible, and so is the reduction of the mean Nu at 100% Nd. A displacement of the traces towards the tip is observed for all jets after the third. Most important, in front of the second jet hole a hot (low Nu) spot is generated, which is adjacent to a cold (high Nu) spot.

The effect of rotation on flow with heat transfer is strongly non-linear and three dimensional. A examination of the (simplified) flow on different planes provides a means for
<table>
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<th>Pc (N/sqm)</th>
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<th>P3</th>
<th>P5</th>
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**Symbols:**
- I/O = IN/OUT-FLOW OF 12TH JET
- Y/N = 12TH JET TRACE APPEARS ON CONTOUR PLOTS? YES/NO
a qualitative explanation of the observed patterns of the jet traces. Consider first Fig.57, which shows the top view of the blade model. The rotational velocity vector \( \vec{\omega} \) is resolved in two components: one, \( \vec{\omega}_p \), parallel to the jet centerlines and the other, \( \vec{\omega}_n \), normal to them. Next consider Fig.58, which shows a cross section through the centerline of the model. A Coriolis acceleration of magnitude \(-2 \vec{\omega}_n \times \vec{v}\) tends to deflect each jet towards the hub. Since the jet emerges in a hotter cross flow, a buoyancy acceleration equal to \(-\vec{\omega}_x (\vec{\omega} \times \vec{r}) \Delta T/T\) tends to deflect it towards the tip and so does the pressure gradient of the cross flow. It is the balance of these principal forces that determines the flow pattern of the jet. Observation of the experimental data supports the above argument:

a) The first jet, at the root of the model, is subject to minimal crossflow and buoyancy effects. Thus Coriolis acceleration dominates and deflects it towards the hub. This explains why the trace of the first jet does not appear on any of the contour plots, although the pressure measurements suggest that it should. It even explains the severe heat transfer gradients, which appear at the root of the model at 180% N\(_d\) (4000 RPM.) At this highest speed, the Coriolis effect dominates on the second jet and deflects it towards the hub. Since the third jet is deflected towards the tip, the in between area of the model skin is uncooled and is adjacent to a region which is cooled very intensely by the second jet. In a real turbine blade, this situation generates large thermal
stresses in a region which is already highly stressed mechanically and can lead to a premature blade failure. An initial cross flow, introduced at the blind end or slanted holes for the first few jets could alleviate this problem.

b) All jets after the third are deflected towards the tip under the influence of increasing cross flow and buoyancy.

Consider next Fig. 59, which shows one of the jets of velocity \( \mathbf{V} \) impinging normally on the wall. In the absence of crossflow and body forces, the jet would spread with cylindrical symmetry and form a sheet of fluid of constant velocity \( \mathbf{V} \) (outside the boundary layer) and thickness inversely proportional to the distance from the stagnation point. If cross flow and body forces are added, the upper half of the sheet is stabilized on the wall by the pressure gradient of the cross flow and the action of the Coriolis force \( -2 \mathbf{m} \times \mathbf{V} \). Furthermore, the fluid particles are accelerated into the page (or, for an actual blade, towards the pressure side) under the influence of the Coriolis \( -2 \mathbf{m} \times \mathbf{V} \). The opposite are true for the part of the fluid sheet flowing towards the hub. Therefore, if Coriolis accelerations are predominant, the jet trace must be distorted from circular to a shape resembling the one of Fig. 59c.

Once more, the experimental data support this argument. The lower jets (second or third to seventh) are deflected towards the trailing side of the model skin. The "spill" towards the leading side is visible for the third jet at the 100% Nd (2000 RPM) contour plots.
CHAPTER 10

CONCLUSIONS

A parametric study of the influence of rotation effects on impingement cooling has been conducted. The investigation included heat transfer measurements on the leading edge of a rotating blade model, under rigorously simulated turbine operating conditions, as well as temperature and pressure measurements inside the cooling gas channels. The spatial resolution achieved through an original radiometric heat transfer measurement technique, was sufficient to observe the traces of individual cooling jets and allowed the generation of Nusselt number (Nu) contour plots, applicable to blade cooling design. The parameters varied were jet mass flow, wall to coolant temperature ratio and rotational speed. The measurements demonstrate, at least for the configuration and the range of parameters tested, that rotational effects are very important and can cause premature blade failure.

Although the accuracy of the results presented here could be improved, at low rotational speeds the data agree well with previously published measurements conducted under stationary conditions. At higher speeds, a monotonic decrease in the average Nu, on the order of 30% is observed at nominal jet Re = 17000 consistent with the observations of Moore and Ayhan [23] and attributed to the counteracting effect of buoyancy on forced convection. A configuration with radial
exhaust at the blade hub, could profit from the this effect.

At higher jet Reynolds numbers Nu displays a minimum, on the order of 30% below the low speed level, from which it recovers as rotational speed and wall to coolant temperature ratio increase. This effect is attributed to buoyancy induced flow reversal in the boundary layer, which promotes mixing and enhances heat transfer.

Severe gradients are generated near the hub region due to the interaction of buoyancy, Coriolis and cross flow forces, which deflect the impingement jets radially inward near the hub and radially outward near the tip. In the intermediate region, a hot spot with adjacent cold spots is generated. The resultant thermal stresses could have detrimental effects on blade life. The reduction of stagger angle to zero, the introduction of initial crossflow or the use of slanted holes for the first three jets are suggested as solutions to this problem.

The radiometric technique developed and the versatile facility built provide powerful experimental tools for further study. Effects of stagger angle, and jet to wall distance can be studied with no modifications on the existing hardware. Other parametric studies and configurations, such as jet hole size and spacing variation, chordwise coolant exhaust and variable heat input per unit area require minor changes of the blade model. Major modifications could allow simulation and/or verification of complete internal blade cooling schemes.
The existing data provides already an overall, qualitative understanding of the flow field and identifies the basic and controlling factors. This, hopefully, will lead in a subsequent effort to correlation, modeling and prediction, analytically or computationally, of the above effects.
REFERENCES

LEADING EDGE IMPINGEMENT COOLING


MIDCHORD IMPINGEMENT COOLING


GENERAL REFERENCES


TEXTBOOKS


EFFECTS OF ROTATION ON HEAT TRANSFER


SECONDARY FLOWS INDUCED BY ROTATION


EFFECTS OF ROTATION ON BOUNDARY LAYER STABILITY


INFRARED DETECTORS
[33] The Infrared Brochure, Santa Barbara Research Center No 73CM.
Fig. 1: MIT-GTL Impingement Cooling Facility
1. Vacuum Chamber
2. Radiometer Housing
3. Instrumentation Rack
4. Vacuum Pump
5. R-502 Compressor
Fig. 2: Rotor
1. Supporting Structure
2. Calibrating Body
3. Heat Exchangers
4. IR Radiometer
5. Power Slip-Rings
Fig. 3: Internal Structure of the Rotor
1. Instrumentation Block
2. Instrumentation Strut
3. Pressure Scanning Valve
4. Power Conductors

Fig. 4: Instrumentation Block
1. Isothermal Block
2. Instrumentation Strut
3. Pressure Tube Splices
Fig. 5: Infrared Radiometer and Translation Stages
1. IR Detector Dewar
2. Imaging System
3. R-Stage
4. Z-Stage

Fig. 6: Imaging System
1. Primary Mirror
2. Secondary Mirror
Fig. 7: Blade Model (Radiometer View)
1. Model Skin
2. Cover
3. Clamp
4. Fixed Flange
5. Moving Flange
6. Power Conductor

Fig. 8: Calibrating Body (Radiometer View)
Fig. 9: Pressure Scanning Valve
1. Valve Body
2. Plunger
3. Pressure Tubes
4. Pushrod
Fig.10: Rotational Encoder and Scanning Valve Actuator
1. Rotational Encoder
2. Slave Hydraulic Cylinder
3. Thrust Bearing
4. Linear Potentiometer
5. Coupling
Fig. 11: Shaft Mounted H. Exchangers and Fixed Flange
1. Heat Exchanger
2. Fixed Flange
3. Inlet Plenum
4. Outlet Plenum
5. Electrical Contact Cavity
Fig. 12: Heat Exchanger Components
1. Bulkhead
2. Tubes
3. Shell
4. End Cover
5. Spacer
Fig. 13: Impingement Cooling Configurations
1. Coolant
2. Impingement Insert
(a) L. Edge Impingement Cooling with Chordwise Exhaust
(b) Midchord Impingement Cooling with Chordwise Exhaust
(c) L. Edge Impingement Cooling with Radial Exhaust

Fig. 14: Fundamental Experimental Concept

Fig. 15: Rotating Coordinate System
Fig. 16: Finite Element Computational Scheme for $h$

\[
h = \frac{\frac{l^2 R_f}{A_f} + \frac{kt}{s_1 s_2} \sum_{i=1}^{4} \frac{|T_i - T_0|}{D_i} + \varepsilon \sigma (T_d - T_0)}{T_0 - T_c}
\]
Fig.17: Experimental Apparatus Schematic Drawing

1. Blade Model
2. Vacuum Chamber
3. Imaging System
4. IR Detector
5. Heat Exchanger
6. Encoder
7. Power Slip Rings
8. Gun Bored Shaft
9. Seal
10. R-12 Inlet
11. R-12 Outlet
12. Var. Speed Drive
13. Instr. Slip Rings
14. Calibration Body
Fig. 18: Blade Model Cross Section (Side View)
1. T = Thermocouple
2. P = Pressure Tap
Fig. 19: Test Section Geometry (Top View)
1. Supply Plenum
2. Jet Hole
3. Impingement Space
4. Impingement Insert
5. Rubber Seal
6. Cover
7. Resistive Wall
Fig. 20: Cooling System Schematic Drawing
1. Rotary Seal (Inlet)
2. Gun Bored Shaft
3. Shaft Mounted H. Exchanger
4. Supply Plenum
5. Exhaust Plenum
6. Rotary Seal (Exhaust)
7. Precooler
8. Economizer
9. Condenser
10. Subcooler
11. Gear Pump
12. Flowmeter
13. Evaporator
Fig. 21: Infrared Radiometer Schematic Drawing
1. IR Detector
2. Primary Mirror
3. Secondary Mirror
4. Model Surface

Fig. 22: Typical HgCdTe IR Detector Performance Curves
1. Range of Spectral Detectivity
2. Detectivity vs. Frequency
3. Responsivity and Noise vs. Frequency
**Fig. 23:** Pressure Scanning Valve Schematic Drawing

1. Pressure Transducer
2. Plunger
3. Valve Body
4. End Cover
5. Stainless Steel Ring
6. O-Ring

**Fig. 24:** Radiometer Repeatability Check
Fig. 25: Standard Deviation of Radiometer Measurements

Fig. 26: Radiometer Voltage to Temperature Conversion
Fig. 27: Computer Controlled Data Acquisition System
Fig. 29: Model Wall Division to Strips
Strips 1-7: Full Span Strips
Strips 8-14: Part Span Strips

Fig. 30: Radiation Intensity Contours - Uncooled Model
Fig. 31: Repeatability Checks - Nu Distribution Plots
Fig. 32: Resettability Checks - Nu Variation Graphs
Fig. 37: T (ok) Distribution for Re=74000, Tw/Tc=Low
Fig. 38: T (°K) Distribution for Re=74000, Tw/Tc=High
Fig. 43: Nu Distribution for Re=74000, Tw/Tc=Low
**Fig. 44:** Nu Distribution for Re=74000, Tw/Tc=High
Fig. 45: 1-D/3-D h Reduction Scheme Comparisons
(a) Low Re, Low Rotational Speed
(b) Medium Re, Med Rotational Speed
Fig. 45: 1-D/3-D h Reduction Scheme Comparisons (c) High Re, High Rotational Speed
Fig. 46: Test#37: First Complete Model Scan
    (a) T (oK) Distribution
    (b) Nu Distribution
Fig. 47: Surface Average Nu - Re Correlation

Fig. 48: Stagnation Strip Nu - Re Correlation
ST. STRIP NU-RE CORRELATION: ALL TESTS

INCREASING BUCYANCY EFFECT

Fig. 56: Stagnation Strip Nu Variation with Buoyancy
Fig. 51: Chordwise Variation of Nu

Fig. 52: Spanwise Variation of Nu
Fig. 53: Forces due to Rotation

Fig. 54: Rotating Channel Flow
Fig. 55: Effect of Buoyancy on Boundary Layer

Fig. 56: Effect of Rotation on Local Nu Distribution
Fig. 57: Resolution of $\vec{\Omega}$

Fig. 58: Jet Flow on a Plane Through the Jet Axis
Fig. 59 Jet Flow on a Plane Normal to the Jet Axis
APPENDICES
APPENDIX 1

CONVENTIONS - SYMBOLS

This Appendix clarifies the various conventions and symbols used for data presentation. Examples are given for some cases.

Al.1 Variable number: VAR = 41

Ø1 - CURRENT RECORD # 23 - MODEL 1st meas( V)
Ø2 - SC. SEQUENCE INDEX 24 - MODEL 2nd meas( V)
Ø3 - RADIUS # (J) 25 - MODEL 3rd meas( V)
Ø4 - THETA # (I) 26 - MODEL 4th meas( V)
Ø5 - THETA COORD (rad) 27 - R.LVL 1st meas( V)
Ø6 - R COORD (m) 28 - R.LVL 2nd meas( V)
Ø7 - Z COORD (m) 29 - R.LVL 3rd meas( V)
Ø8 - X FOLDED COORD. (m) 30 - R.LVL 4th meas( V)
Ø9 - Y COORD. (m) 31 - BODY 1st meas( V)
1Ø - X UNFOLD.COORD. (m) 32 - BODY 2nd meas( V)
11 - PX POSITION (V) 33 - BODY 3rd meas( V)
12 - TC M. SKIN/HUB (ok) 34 - BODY 4th meas( V)
13 - TC B. SKIN/TIP (ok) 35 - MODEL MEAN (V)
14 - TC B. SKIN/HUB (ok) 36 - MODEL S.DEV. (V)
15 - TC PLENUM/HUB (ok) 37 - REP. LVL MEAN (V)
16 - TC PLENUM/TIP (ok) 38 - REP. LVL S.DEV.(V)
17 - TC IMP.CH./HUB (ok) 39 - BODY MEAN (V)
18 - TC IMP.CH./TIP (ok) 40 - BODY S.DEV. (V)
19 - AD59Ø R. TEMP.(ok) 41 - SPOT TEMP. (ok)
2Ø - MODEL CURRENT (A) 42 - SPOT h (W/sq m ok)
21 - PRESS. @ PX (N/m2) 43 - SPOT Nu
22 - S. RING CURRENT(A) 44 - (Reserved)

Al.2 File code: DR2:GRØ37L.TST

First 4 digits : Storage device (may be omitted) ....... DR2:
Next 2 " : Letter code number......................... GR
" 3 " : Test serial number........................Ø37
" 1 " : Local (L) or average (A) data............. L
Last 4 " : File type.............................. TST

Al.3 Temperature ratios: Tl = 1.333

Tl = Temperature of supply coolant * 1.333
(Mean of all measurements in ok)

Tl = TC at blade model skin hub
T2 = " " calibrating body " tip
T3 = " " " " hub
T4 = " " coolant supply plenum hub
T5 = " " " " tip
$T_6 = " \text{impingement channel at hub}"

$T_7 = " \text{tip}"

$T_8 = \text{Reference temperature (AD590 M)}$

$T_9 = \text{Average model T at imping. cooled span} $

$T_{10} = \text{skin surface temperature} \ $

Al.4 Pressure ratios: $P_2 = 0.939$

$P_2 = \text{Pressure of supply coolant} \times 0.939$

(Mean of all measurements in N/sq m abs.)

$P_1 = \text{Supply plenum at hub}$

$P_2 = \text{Exhaust at tip}$

$P_3 = \text{Supply at tip}$

$P_4 = \text{Impingement channel at hub}$

$P_5 = \text{Vacuum chamber (reference)}$

Al.5 Average Nu: $7 = 0.812E+02$

The blade model surface is divided in seven (7) imaginary, equal width, radial strips (see Fig.29.) The first strip starts at $X = \theta.\theta$ and the last ends at $X = X_{\text{max}}$ on the unfolded skin. Each strip is divided in two sections: one corresponding to the part of span cooled by the impinging jets, and another, corresponding to the rest of the span which is cooled by convection. Thus, strips #1 to #7 correspond to the full skin span, while strips #8 to #14 to the impingement cooled span. The "stagnation" strips correspond to #4 and #11.

Average Nu are computed for each strip, according to the scheme of Sect.6.4. The first seven averages correspond to the full - span strips. The next seven to part span strips. The "stagnation" strips correspond to numbers 4 and 11. The last Nu is the average of the whole surface. Zero Nu means that no measurements were made on the strip in question, because it is not on the line of sight of the radiometer.
APPENDIX 2

PRESSURE CORRECTIONS

A2. 1 Introduction.

As described in Sect.4. 3, pressure measurements are taken inside the coolant passages of the model blade, through a shaft mounted pressure scanning valve. Pressure is transmitted through small diameter (0.8 mm (0.032 in.) stainless steel pressure tubes. The raw pressure reading from the pressure transducer (KULITE model CQ-080-50) is corrected for DC offset and centrifugal effects according to the following method.

A2. 2 DC Offset Compensation.

The pressure inside the vacuum chamber is measured through the pressure transducer when the plunger measuring orifice is located at the last port of the pressure scanning valve. This measurement is compared to the one taken through the vacuum gauge and their difference is set equal to the pressure measurement system DC offset compensation for thermal drift and centrifugal effects on the pressure transducer diaphragm.

A2. 3 Centrifugal Effect Compensation.

The equilibrium condition for every cross section of a pressure tube filled with a perfect gas is:
\[ A \, dp = A \, dr \, \Omega^2 \, r \]  
(Equ.A2.1)

\[ dp = \frac{p}{RT} \, \Omega^2 \, r \, dr \Rightarrow \int \frac{dp}{p} = \int \frac{\Omega^2 \, r}{RT} \, dr \]  
(Equ.A2.2)

\[ \ln \frac{p}{px} = \frac{\Omega^2}{R} \int \frac{r \, dr}{T} \]  
(Equ.A2.3)

Therefore corrections can be made if the temperature distribution along the pressure tubes is known.

The pressure tubes are brazed side by side on the instrumentation strut (see Sect.4.2 and Fig. 4,) which provides a torsionally flexible but radially rigid support. This strut is insulated by the vacuum of the chamber, and has no heat sources. Therefore, the temperature distribution along its length is linear, due to thermal conduction from one end to the other. The (cold) shaft end is mounted on the instrumentation block and therefore its temperature is the measured reference temperature. The (hot) blade model end is at the root of the instrumentation spine.

Some of the pressure tubes continue encapsulated inside the instrumentation spine (Fig.18,) which is considered isothermal for pressure correction purposes, due to its relatively large cross section. Its constant temperature is taken equal to the mean of the readings of the two thermocouples at coolant supply plenum hub and tip.
Therefore for the intrumentation strut:

\[ T(r) = a \frac{r}{px} + b \quad \text{for} \quad r < r < r_{\text{hub}} \quad \text{(Equ. A2.4)} \]

with:

\[ T = T_{\text{ref}} \quad \text{for} \quad r = r_{\text{px}} \]

\[ T = \left( \frac{T_{\text{hub}} + T_{\text{tip}}}{2} \right) \quad \text{for} \quad r = r_{\text{hub}} \]

which yields:

\[ a = \left( \frac{T_{\text{hub}} + T_{\text{tip}}}{2} \right) / \left( r_{\text{ref}} - r_{\text{px}} \right) \quad \text{(Equ. A2.5)} \]

\[ b = T_{\text{ref}} - a \frac{r}{px} \quad \text{(Equ. A2.6)} \]

and for the intrumentation spine:

\[ T(r) = \left( \frac{T_{\text{hub}} + T_{\text{tip}}}{2} \right) = \text{constant} \quad \text{(Equ. A2.7)} \]

\[ \text{for} \quad r > r_{\text{hub}} \]

Returning to Equ. A2.4:

\[ \ln \frac{p}{p_{\text{px}}} = \frac{\Omega^2}{R} \left( \left[ \frac{r}{a} - \frac{b}{2} \ln(\frac{ar + b}{a}) \right]_{\text{hub}}^{r_{\text{hub}}} + \left[ \frac{2}{2T_{\text{avg}}} \frac{r}{r_{\text{hub}}} \right] \right) \quad \text{(Equ. A2.8)} \]

\[ p = p_{\text{px}} \exp \left[ \frac{\Omega^2}{R} \left( \left[ \frac{r}{a} - \frac{b}{2} \ln(\frac{ar + b}{a}) \right]_{\text{hub}}^{r_{\text{hub}}} + \left[ \frac{2}{2T_{\text{avg}}} \frac{r}{r_{\text{hub}}} \right] \right) \right] \quad \text{(Equ. A2.9)} \]

Equ. A2.9 is used for centrifugal effect pressure correction.

A2. 3 Pressure Corrections.

The absolute pressure at the tap radius is computed from the raw pressure measurement as follows:
\[ p = \left( p_{\text{abs}} - p_{\text{raw}} - p_{\text{off}} \right) CF(r_{\text{ref}}, T_{\text{tap}}, T_{\text{ref}}, T_{\text{hub}}, T_{\text{tip}}) \quad (A2.10) \]
APPENDIX 3

MEAN AND NON - DIMENSIONAL PARAMETERS

(ALL TESTS)
TEST # 35 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:  
# OF JETS = 12  JET DIA (m) = $9.296E-02$
SUPPLY PLENUM AREA (sq. m) = $9.130E-03$
" PERIMETER (m) = $9.498E-01$
ST. ANGLE= 30.00 HUB/TIP= 0.800 MEAN RADIUS= 5.457m
s/d = 2.0  p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 9.8
FLOWMETER TEMPERATURE (oC) = 255.4
(C) = 0.8
CORRECTED R-12 MASS FLOW (kg/sec) = $9.0000$
REFERENCE PRESSURE (mm Hg) = 9.200
(N/sq. m.) = $9.266E+02$
SHAFT SPEED (rad/sec) = 76.0
(RPM) = 726.
MODEL SUPPLY CURRENT (Amperes) = 20.84
HEAT LOAD/UNIT AREA (W/sq m) = $9.7367E+04$

TEMPERATURE RATIOS: T.S. COOLANT (oC) = 317.1
T1= 1.177 T2= 0.951 T3= 0.951 T4= 1.000 T5 = 1.033
T6= 1.125 T7= 1.041 T8= 0.936 T9 = 1.584 T10= 1.574

PRESSURE RATIOS: P.S. COOLANT (N/sq.m.) = $9.272E+02$
P1= 1.000 P2= 1.000 P3= 1.810 P4= 1.001 P5= 1.812 P6= 0.938

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m.) = $9.133E-04$
THERMAL CONDUCTIVITY k (W/m. oK) = $9.106E-01$
Cp (J/kg oK) = 618.7
Cp/Cv = 1.125
Pv = 0.7755

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE= $9.000  IMPINGEMENT JET= $0.000$

NONDIMENSIONAL PARAMETERS:
Re Supply = $9.000E+08  Re Jet = $0.000E+00$
Ro " = $9.000E+08  Ro " = $0.000E+00$
M " = $9.000E+08  M " = $9.000E+00$

AVERAGE NUSSELT NUMBERS:
1= $9.000E+08  2= 9.800E+08  3= 0.179E+08  4= 0.862E+01$
5= $9.388E+01  6= 0.189E+02  7= 0.277E+02  8= 0.000E+00$
9= $9.000E+08  10= 0.172E+02  11= 0.811E+01  12= 0.173E+01$
13= $9.198E+02  14= 0.271E+02  15= 0.130E+02$
TEST # 37 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m) = 0.130E-03
" PERIMETER (m) = 0.493E-01
ST. ANGLE= 30.0° HUB/TIP= 0.800 MEAN RADIUS= 0.457m
s/d = 2.9 p/d = 2.963 D/d = 5.986

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 15.3
FLOWMETER TEMPERATURE (°C) = 263.9
(oF) = 15.4
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00746
REFERENCE PRESSURE (mm Hg) = 0.300
(N/sq. m.) = 0.3999E+02
SHAFT SPEED (rad/sec) = 88.2
(kn) = 843.
MODEL SUPPLY CURRENT (Ampere) = 33.36
HEAT LOAD/UNIT AREA (W/sq m) = 0.2496E+05

TEMPERATURE RATIOS: T S. COOLANT (°C) = 321.4
T1= 1.210 T2= 0.971 T3= 0.973 T4= 1.000 T5= 1.009
T6= 1.033 T7= 1.027 T8= 0.927 T9= 1.305 T10= 1.346

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = 0.102E+06
P1= 1.000 P2= 0.940 P3= 1.023 P4= 0.973 P5= 0.967 P6= 0.000

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m.) = 0.134E-04
THERMAL CONDUCTIVITY k (W/m. °C) = 0.108E-01
CP (J/kg °C) = 622.6
CP/Cv = 1.1241
Pr = 0.7738

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE= 12.506 IMPINGEMENT JET= 41.312

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.1113E+05 Re Jet = 0.2861E+05
Ro " = 0.6899E+02 Ro " = 0.2276E+03
M " = 0.7935E-01 M " = 0.2627E+00

AVERAGE NUSSELT NUMBERS:
1= 0.000E+00 2= 0.000E+00 3= 0.245E+03 4= 0.180E+03
5= 0.137E+03 6= 0.112E+03 7= 0.109E+03 8= 0.000E+00
9= 0.000E+00 10= 0.279E+03 11= 0.219E+03 12= 0.155E+03
13= 0.124E+03 14= 0.115E+03 15= 0.151E+03
TEST # 39 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" PERIMETER (m) = 0.498E+01
ST. ANGLE = 30.00 HUB/TIP = 0.800 MEAN RADIUS = 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 19.8
FLOWMETER TEMPERATURE (°C) = 258.7
(°F) = 6.0
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00910
REFERENCE PRESSURE (mm Hg) = 0.460
(N/sq. m.) = 0.5332E+02
SHAFT SPEED (rad/sec) = 37.7
(RPM) = 837.
MODEL SUPPLY CURRENT (Amperes) = 46.15
HEAT LOAD/UNIT AREA (W/sq m) = 0.3612E+05

TEMPERATURE RATIOS: T S. COOLANT (°C) = 323.2
T1 = 1.259 T2 = 0.985 T3 = 0.988 T4 = 1.000 T5 = 1.009
T6 = 1.042 T7 = 1.030 T8 = 0.919 T9 = 1.322 T10 = 1.359

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = 0.120E+06
P1 = 1.000 P2 = 0.942 P3 = 1.004 P4 = 0.927 P5 = 0.933 P6 = 0.000

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY \( \mu \) (Nsec/sq.m.) = 0.135E-04
THERMAL CONDUCTIVITY \( k \) (W/m. °C) = 0.109E+01
\( \text{Cp} \) (J/kg °C) = 624.3
\( \text{Cp/Cv} \) = 1.1237
Pr = 0.7731

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 13.822 IMPINGEMENT JET = 45.268

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.1441E+05 Re Jet = 0.3703E+05
Ro " = 0.7661E+02 Ro " = 0.2509E+03
M " = 0.8746E+01 M " = 0.2872E+00

AVERAGE NUSSELT NUMBERS:
1 = 0.00E+00 2 = 0.00E+00 3 = 0.319E+03 4 = 0.231E+03
5 = 0.176E+03 6 = 0.143E+03 7 = 0.133E+03 8 = 0.000E+00
9 = 0.000E+00 10 = 0.358E+03 11 = 0.279E+03 12 = 0.196E+03
13 = 0.155E+03 14 = 0.143E+03 15 = 0.193E+03
TEST # 40 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:
- # OF JETS = 12
- JET DIA (m) = $0.206E-02$
- SUPPLY PLENUM AREA (sq. m) = $0.138E-03$
- PERIMETER (m) = $0.498E-01$
- ST. ANGLE = 38.8° HUB/TIP = 0.800 MEAN RADIUS = 0.457m
- s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA:
- VOLUME FLOW FROM FLOWMETER (% of max) = 39.7
- FLOWMETER TEMPERATURE (°K) = 255.8
- (°F) = 7.7
- CORRECTED R-12 MASS FLOW (kg/sec) = $0.01954$
- REFERENCE PRESSURE (mm Hg) = $0.446$
- (N/sq. m) = $0.532E+02$
- SHAFT SPEED (rad/sec) = 96.3
- (RPM) = 863.
- MODEL SUPPLY CURRENT (Amperes) = 50.46
- HEAT LOAD/UNIT AREA (W/sq m) = $0.431E+05$

TEMPERATURE RATIOS:
- T.S. COOLANT (°K) = 323.4
- T1 = 1.258 T2 = 0.999 T3 = 1.003 T4 = 1.005 T5 = 1.006
- T6 = 1.036 T7 = 1.019 T8 = 0.928 T9 = 1.282 T10 = 1.315

PRESSURE RATIOS:
- P.S. COOLANT (N/sq.m) = $0.214E+06$
- P1 = 1.000 P2 = 0.868 P3 = 0.988 P4 = 0.927 P5 = 0.887 P6 = 0.800

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
- VISCOSITY $\mu$ (Nsec/sq.m) = $0.135E-04$
- THERMAL CONDUCTIVITY $k$ (W/m. °K) = $0.109E-01$
- $C_p$ (J/kg °K) = 624.5
- $C_p/C_v$ = 1.1237
- $Pr$ = $0.7736$

COOLANT VELOCITIES (m/sec):
- SUPPLY PASSAGE = 15.681
- IMPINGEMENT JET = 50.693

NONDIMENSIONAL PARAMETERS:
- Re Supply = $0.2899E+05$
- Re Jet = $0.7451E+05$
- Ro " = $0.8437E+02$
- Ro " = $0.2727E+03$
- $M$ = $0.9920E-01$
- $M$ " = $0.3217E+00$

AVERAGE NUSSELT NUMBERS:
- 1= $0.000E+00$
- 2= $0.200E+00$
- 3= $0.558E+03$
- 4= $0.415E+03$
- 5= $0.312E+03$
- 6= $0.231E+03$
- 7= $0.191E+03$
- 8= $0.000E+00$
- 9= $0.000E+00$
- 10= $0.635E+03$
- 11= $0.509E+03$
- 12= $0.355E+03$
- 13= $0.256E+03$
- 14= $0.205E+03$
- 15= $0.332E+03$
TEST # 41 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:  
# OF JETS = 12  JET DIA (m) = 0.206E-02  
SUPPLY PLENUM AREA (sq. m.) = 0.138E-03  
" PERIMETER (m.) = 0.498E-01  
ST. ANGLE= 38.8°  HUB/TIP= 0.800  MEAN RADIUS= 0.457m  
s/d = 2.0  p/d = 2.963  D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 40.6  
FLOWMETER TEMPERATURE (°C) = 255.6  
(OF) = 0.5  
CORRECTED R-12 MASS FLOW (kg/sec) = 0.51998  
REFERENCE PRESSURE (mm Hg) = 0.350  
(N/sq. m.) = 0.4665E+02  
SHAFT SPEED (rad/sec) = 94.6  
(RPM) = 983.  
MODEL SUPPLY CURRENT (Amperes) = 61.50  
HEAT LOAD/UNIT AREA (W/sq. m) = 0.6415E+05

TEMPERATURE RATIOS: T S. COOLANT (°C) = 332.0  
T1= 1.339  T2= 1.001  T3= 1.008  T4= 1.005  T5= 1.009  
T6= 1.048  T7= 1.027  T8= 0.916  T9= 1.335  T10= 1.378

PRESSURE RATIOS: P S. COOLANT (N/sq. m.) = 0.2126E+06  
P1= 1.000  P2= 0.901  P3= 1.019  P4= 0.930  P5= 0.916  P6= 0.000

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:  
VISCOSITY u (Nsec/sq.m.) = 0.138E-04  
THERMAL CONDUCTIVITY k (W/m. °C) = 0.114E-01  
Cp  
(J/kg OK) = 632.1  
Cp/Cv = 1.1220  
Pr = 0.7696

COOLANT VELOCITIES (m/sec):  
SUPPLY PASSAGE= 16.573  IMPINGEMENT JET= 53.276

NONDIMENSIONAL PARAMETERS:  
Re Supply = 0.2893E+05  Re Jet = 0.7436E+05  
Ro " = 0.8514E+02  Ro " = 0.2737E+03  
M " = 0.1035E+00  M " = 0.3340E+00

AVERAGE NUSSELT NUMBERS:  
1= 0.000E+00  2= 0.000E+00  3= 0.495E+03  4= 0.361E+03  
5= 0.281E+03  6= 0.228E+03  7= 0.189E+03  8= 0.000E+00  
9= 0.000E+00  10= 0.556E+03  11= 0.436E+03  12= 0.317E+03  
13= 0.251E+03  14= 0.204E+03  15= 0.302E+03
TEST # 42 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: 
- # OF JETS = 12  
- JET DIA (m) = $0.206E-02$  
- SUPPLY PLENUM AREA (sq. m) = $0.130E-03$  
- " PERIMETER (m) = $0.498E-01$  
- ST. ANGLE= 30.00 HUB/TIP= 0.800 MEAN RADIUS= 0.457m  
- s/d = 2.0  p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 9.6  
- FLOWMETER TEMPERATURE (oK) = 269.9  
- (oF) = 26.2
- CORRECTED R-12 MASS FLOW (kg/sec) = $0.80471$  
- REFERENCE PRESSURE (mm Hg) = $0.300$  
- (N/sq. m.) = $0.399E+02$  
- SHAFT SPEED (rad/sec) = 91.0  
- (RPM) = 859.
- MODEL SUPPLY CURRENT (Amperes) = 37.52  
- HEAT LOAD/UNIT AREA (W/sq m) = $0.2388E+05$

TEMPERATURE RATIOS: T S. COOLANT (oK) = 321.6
- T1 = 1.236  T2 = 0.966  T3 = 0.968  T4 = 1.000  T5 = 1.015  
- T6 = 1.041  T7 = 1.042  T8 = 0.918  T9 = 1.357  T10 = 1.393

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = $0.673E+05$
- P1 = 1.000  P2 = 0.934  P3 = 0.982  P4 = 0.981  P5 = 0.931  P6 = 0.001

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
- VISCOSITY $u$ (Nsec/sq.m.) = $0.135E-04$  
- THERMAL CONDUCTIVITY k (W/m. oK) = $0.106E-01$  
- $C_p$ (J/kg oK) = 622.8
- $C_p/C_v = 1.1241$  
- Pr = $0.7737$

COOLANT VELOCITIES (m/sec):
- SUPPLY PASSAGE= 11.929  IMPINGEMENT JET= 39.541

NONDIMENSIONAL PARAMETERS:
- Re Supply = $0.7018E+04$  Re Jet = $0.1804E+05$  
- Ro " = $0.6372E+02$  Ro " = $0.2112E+03$  
- M " = $0.7567E-01$  M " = $0.2513E+00$

AVERAGE NUSSELT NUMBERS:
1= $0.000E+00$  2= $0.000E+00$  3= $0.155E+03$  4= $0.118E+03$  
5= $0.884E+02$  6= $0.796E+02$  7= $0.318E+02$  8= $0.000E+00$  
9= $0.000E+00$  10= $0.172E+03$  11= $0.141E+03$  12= $0.980E+02$  
13= $0.857E+02$  14= $0.856E+02$  15= $0.181E+03$
TEST # 43 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) =\.206E-02
SUPPLY PLENUM AREA (sq. m) =\.130E-03
  " PERIMETER (m) =\.498E-01
ST. ANGLE = 30.0° HUB/TIP =\.800 MEAN RADIUS =\.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 10.0
FLOWMETER TEMPERATURE (°C) = 268.0
  (°F) = 22.7
CORRECTED R-12 MASS FLOW (kg/sec) =\.00488
REFERENCE PRESSURE (mm Hg) =\.300
  (N/sq. m) =\.3999E+02
SHAFT SPEED (rad/sec) = 90.8
  (RPM) = 867.
MODEL SUPPLY CURRENT (Amperes) = 31.9
HEAT LOAD/UNIT AREA (W/sq m) =\.1726E+05

TEMPERATURE RATIOS: T S. COOLANT (°C) = 325.0
T1 = 1.179 T2 =\.950 T3 =\.951 T4 = 1.000 T5 = 1.014
T6 = 1.032 T7 =\.030 T8 =\.916 T9 = 1.296 T10 = 1.325

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) =\.834E+05
P1 = 1.000 P2 =\.977 P3 = 1.020 P4 =\.989 P5 =\.998 P6 =\0.000

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY \( \mu \) (Nsec/sq.m.) =\.136E-04
THERMAL CONDUCTIVITY \( k \) (W/m. °C) =\.110E-01
\( \text{C}_p \)
  (J/kg °C) = 625.9
\( \text{C}_p / \text{C}_v \)
  = 1.1234
Pr
  =\.7724

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 10.879 IMPINGEMENT JET = 33.765

NONDIMENSIONAL PARAMETERS:
Re Supply =\.7200E+04 Re Jet =\.1851E+05
Ro " =\.5394E+02 Ro " =\.1897E+03
M " =\.6361E-01 M " =\.2134E+00

AVERAGE NUSSELT NUMBERS:
1 =\.000E+00 2 =\.000E+00 3 =\.0169E+03 4 =\.130E+03
5 =\.0869E+02 6 =\.843E+02 7 =\.771E+02 8 =\.300E+00
9 =\.000E+00 10 =\.0189E+03 11 =\.156E+03 12 =\.108E+03
13 =\.918E+02 14 =\.800E+02 15 =\.108E+03
TEST # 44 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:

# OF JETS = 12  JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.138E-03
" PERIMETER (m.) = 0.498E-01
ST. ANGLE = 30.80 HUB/TIP = 0.800 MEAN RADIUS = 0.457m
s/d = 2.8  p/d = 2.963 D/d = 5.886

TEST DATA:

VOLUME FLOW FROM FLOWMETER (% of max) = 19.5
FLOWMETER TEMPERATURE (OK) = 256.9
(CF) = 2.7
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00962
REFERENCE PRESSURE (mm Hg) = 0.300
(N/sq. m.) = 0.3999E+02
SHAFT SPEED (rad/sec) = 89.8
(RPM) = 857.
MODEL SUPPLY CURRENT (Ampere) = 50.23
HEAT LOAD/UNIT AREA (W/sq m) = 0.4280E+05

TEMPERATURE RATIOS:

T S. COOLANT (OK) = 334.6
T1 = 1.286 T2 = 0.961 T3 = 0.966 T4 = 1.000 T5 = 1.012
T6 = 1.048 T7 = 1.034 T8 = 0.999 T9 = 1.351 T10 = 1.398

PRESSURE RATIOS:

P S. COOLANT (N/sq.m.) = 0.120E+06
P1 = 1.000 P2 = 0.934 P3 = 1.024 P4 = 0.965 P5 = 0.952 P6 = 0.800

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:

VISCOSITY u (Nsec/sq.m.) = 0.148E-04
THERMAL CONDUCTIVITY k (W/m. OK) = 0.115E-01
Cp = 634.4
Cp/Cv = 1.1215
Pr = 0.7686

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 14.197  IMPINGEMENT JET = 46.419

NONDIMENSIONAL PARAMETERS:

Re Supply = 0.1383E+05  Re Jet = 0.3554E+05
Ro " = 0.7688E+02  Ro " = 0.2514E+03
M " = 0.8839E-01  M " = 0.2897E+00

AVERAGE NUSSELT NUMBERS:

1 = 0.000E+00  2 = 0.000E+00  3 = 0.283E+03  4 = 0.286E+03
5 = 0.156E+03  6 = 0.129E+03  7 = 0.125E+03  8 = 0.000E+00
9 = 0.000E+00  10 = 0.322E+03  11 = 0.251E+03  12 = 0.177E+03
13 = 0.143E+03  14 = 0.133E+03  15 = 0.173E+03
TEST # 54 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" PERIMETER (m.) = 0.498E-01
ST. ANGLE= 30.0° HUB/TIP= 0.800 MEAN RADIUS= 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 20.3
FLOWMETER TEMPERATURE (°K) = 256.2
(°F) = 1.4
CORRECTED R-12 MASS FLOW (kg/sec) = 0.81001
REFERENCE PRESSURE (mm Hg) = 0.250
(N/sq. m.) = 0.333E+02
SHAFT SPEED (rad/sec) = 150.1
(RPM) = 1433.
MODEL SUPPLY CURRENT (Amperes) = 50.53
HEAT LOAD/UNIT AREA (W/sq m) = 0.4332E+05

TEMPERATURE RATIOS: T S. COOLANT (°K) = 332.2
T1 = 1.292 T2 = 0.964 T3 = 0.969 T4 = 1.000 T5 = 1.014
T6 = 1.045 T7 = 1.040 T8 = 0.895 T9 = 1.372 T10 = 1.421

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = 0.116E+06
P1 = 1.000 P2 = 0.934 P3 = 1.032 P4 = 0.966 P5 = 0.983 P6 = 0.800

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m.) = 0.139E-04
THERMAL CONDUCTIVITY k (W/m. °K) = 0.114E-01
Cp (J/kg °K) = 632.2
Cp/Cv = 1.1220
Pr = 0.7696

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE= 15.187 IMPINGEMENT JET= 49.314

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.1448E+05 Re Jet = 0.3722E+05
Ro " = 0.4918E+02 Ro " = 0.1597E+03
M " = 0.9487E-01 M " = 0.3890E+00

AVERAGE NUSSEL NUMBERS:
1= 0.000E+00 2= 0.000E+00 3= 0.247E+03 4= 0.182E+03
5= 0.142E+03 6= 0.125E+03 7= 0.115E+03 8= 0.000E+00
9= 0.000E+00 10= 0.279E+03 11= 0.219E+03 12= 0.159E+03
13= 0.137E+03 14= 0.124E+03 15= 0.157E+03
TEST # 55 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: 
\# OF JETS = 12  
JET DIA (m) = 0.205E-02  
SUPPLY PLENUM AREA (sq. m.) = 0.138E-03  
" PERIMETER (m.) = 0.498E-01  
ST. ANGLE= 30°  HUB/TIP= 0.800 MEAN RADIUS= 0.457m  
s/d = 2.0  p/d = 2.963  D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 19.6
FLOWMETER TEMPERATURE  (OK) = 256.9  
(oF) = 2.8
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00964  
REFERENCE PRESSURE (mm Hg) = 0.250
(N/sq. m.) = 0.3333E+02
SHAFT SPEED (rad/sec) = 148.7  
(RPM) = 1420.
MODEL SUPPLY CURRENT (Amperes) = 45.87
HEAT LOAD/UNIT AREA (W/sq m) = 0.3569E+05

TEMPERATURE RATIOS:  T.S. COOLANT (OK) = 334.6
T1 = 1.251  T2 = 0.949  T3 = 0.952  T4 = 1.000  T5 = 1.013  
T6 = 1.035  T7 = 1.035  T8 = 0.901  T9 = 1.339  T10 = 1.381

PRESSURE RATIOS:  P.S. COOLANT (N/sq.m.) = 0.116E+06
P1 = 1.000  P2 = 0.934  P3 = 1.038  P4 = 0.969  P5 = 1.003  P6 = 0.000

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY \( u \) (Nsec/sq.m.) = 0.140E-04
THERMAL CONDUCTIVITY \( k \) (W/m. OK) = 0.115E-01
\( C_p \) (J/kg OK) = 634.4
\( C_p/C_v \) = 1.1215
Pr = 0.7686

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 14.745  IMPINGEMENT JET = 48.031

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.1385E+05  \( Re \) Jet = 0.3560E+05
Ro " = 0.4821E+02  Ro " = 0.570E+03
M " = 0.9179E-01  M " = 0.2998E+00

AVERAGE NUSSELT NUMBERS:
1 = 0.000E+00  2 = 0.000E+00  3 = 0.241E+03  4 = 0.178E+03
5 = 0.138E+03  6 = 0.122E+03  7 = 0.110E+03  8 = 0.000E+00
9 = 0.000E+00  10 = 0.272E+03  11 = 0.215E+03  12 = 0.155E+03
13 = 0.134E+03  14 = 0.118E+03  15 = 0.152E+03
TEST # 56 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: 
# OF JETS = 12  JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" PERIMETER (m.) = 0.498E-01
ST. ANGLE= 30.00 HUB/TIP = 0.300 MEAN RADIUS = 0.457m
s/d = 2.0  p/d = 2.963  D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 10.0
FLOWMETER TEMPERATURE (oK) = 264.4
(oF) = 16.3
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00488
REFERENCE PRESSURE (mm Hg) = 0.200
(N/sq. m.) = 0.2666E+02
SHAFT SPEED (rad/sec) = 147.6
(RPM) = 1409.
MODEL SUPPLY CURRENT (Amperes) = 37.34
HEAT LOAD/UNIT AREA (W/sq. m) = 0.2365E+05

TEMPERATURE RATIOS: T S. COOLANT (oK) = 336.4
T1 = 1.214  T2 = 9.27  T3 = 0.929  T4 = 1.0E0  T5 = 1.020
T6 = 1.035  T7 = 1.046  T8 = 0.990  T9 = 1.351  T10 = 1.392

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = 0.644E+05
P1 = 1.00  P2 = 9.29  P3 = 1.010  P4 = 0.953  P5 = 0.973  P6 = 0.000

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m.) = 0.140E-04
THERMAL CONDUCTIVITY k (W/m. oK) = 0.116E-01
Cp (J/kg oK) = 635.9
Cp/Cv = 1.1212
Pr = 0.7680

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 13.522  IMPINGEMENT JET = 44.412

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.6983E+04  Re Jet = 0.1795E+05
Ro " = 0.4453E+02  Ro " = 0.1463E+03
M " = 0.8398E-01  M " = 0.2764E+00

AVERAGE NUSSELT NUMBERS:
1 = 0.000E+00  2 = 0.000E+00  3 = 0.138E+03  4 = 0.106E+03
5 = 0.796E+02  6 = 0.751E+02  7 = 0.724E+02  8 = 0.900E+00
9 = 0.000E+00  10 = 0.156E+03  11 = 0.128E+03  12 = 0.891E+02
13 = 0.826E+02  14 = 0.776E+02  15 = 0.910E+02
TEST # 57 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.256E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" " PERIMETER (m.) = 0.498E-01
ST. ANGLE= 30.00 HUB/TIP= 0.800 MEAN RADIUS= 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 10.0
FLOWMETER TEMPERATURE (°C) = 267.6
(°F) = 22.0
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00486
REFERENCE PRESSURE (mm Hg) = 0.250
(N/sq. m.) = 0.3333E+02
SHAFT SPEED (rad/sec) = 147.3
(RPM) = 1487.
MODEL SUPPLY CURRENT (Amperes) = 31.58
HEAT LOAD/UNIT AREA (W/sq m) = 3.1692E+05

TEMPERATURE RATIOS: T.S. COOLANT (°C) = 334.5
T1= 1.168 T2= 0.924 T3= 0.925 T4= 1.000 T5= 1.018
T6= 1.028 T7= 1.036 T8= 0.921 T9= 1.302 T10= 1.333

PRESSURE RATIOS: P.S. COOLANT (N/sq.m.) = 0.781E+05
P1= 1.000 P2= 0.958 P3= 1.028 P4= 0.961 P5= 0.988 P6= 0.900

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m.) = 0.139E-04
THERMAL CONDUCTIVITY k (W/m. °C) = 0.115E-01
Cp (J/kg °C) = 633.9
Cp/Cv = 1.1216
Pr = 0.7639

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE 11.026 IMPINGEMENT JET= 36.747

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.7082E+04 Re Jet = 0.1808E+05
Ro " = 0.3637E+02 Ro " = 0.1212E+03
M " = 0.6875E-01 M " = 0.2293E+00

AVERAGE NUSSELT NUMBERS:
1= 0.000E+00 2= 0.000E+00 3= 0.142E+03 4= 0.108E+03
5= 0.842E+02 6= 0.774E+02 7= 0.594E+02 8= 0.000E+00
9= 0.000E+00 10= 0.159E+03 11= 0.130E+03 12= 0.918E+02
13= 0.848E+02 14= 0.741E+02 15= 0.928E+02
TEST # 58 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" PERIMETER (m.) = 0.498E-01
ST. ANGLE= 30.50 HUB/TIP= 0.800 MEAN RADIUS= 0.457
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 41.7
FLOWMETER TEMPERATURE (oK) = 252.2
OF = -5.8
CORRECTED R-12 MASS FLOW (kg/sec) = 0.02066
REFERENCE PRESSURE (mm Hg) = 0.440
(N/sq. m.) = 0.533E+02
SHAFT SPEED (rad/sec) = 144.7
(RPM) = 1382.
MODEL SUPPLY CURRENT (Amperes) = 60.61
HEAT LOAD/UNIT AREA (W/sq m) = 0.6231E+05

TEMPERATURE RATIOS: T S. COOLANT (oK) = 332.2
T1 = 1.333 T2 = 0.989 T3 = 0.995 T4 = 1.000 T5 = 1.009
T6 = 1.043 T7 = 1.029 T8 = 0.898 T9 = 1.345 T10 = 1.393

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = 0.194E+06
P1 = 1.000 P2 = 0.987 P3 = 1.055 P4 = 0.949 P5 = 0.959 P6 = 0.800

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m.) = 0.139E+04
THERMAL CONDUCTIVITY k (W/m. oK) = 0.114E+01
Cp (J/kg oK) = 632.3
Cp/Cv = 1.1220
Pr = 0.7696

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 18.695 IMPINGEMENT JET = 59.133

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.2981E+05 Re Jet = 0.7662E+05
Ro " = 0.6278E+02 Ro " = 0.1985E+03
M " = 0.1168E+00 M " = 0.3709E+00

AVERAGE NUSSELT NUMBERS:
1 = 0.000E+00 2 = 0.000E+00 3 = 0.426E+03 4 = 0.308E+03
5 = 0.246E+03 6 = 0.207E+03 7 = 0.178E+03 8 = 0.800E+00
9 = 0.000E+00 10 = 0.431E+03 11 = 0.373E+03 12 = 0.278E+03
13 = 0.228E+03 14 = 0.193E+03 15 = 0.255E+03
TEST # 59  TASK # 2:  MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:  # OF JETS = 12  JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.138E-03
" PERIMETER (m.) = 0.498E+01
ST. ANGLE= 30.0° HUB/TIP = 0.800 MEAN RADIUS = 0.457m
s/d = 2.0  p/d = 2.963  D/d = 5.886

TEST DATA:  VOLUME FLOW FROM FLOWMETER (% of max) = 40.5
FLOWMETER TEMPERATURE (oK) = 252.0
(oF) = -6.0
CORRECTED R-12 MASS FLOW (kg/sec) = 0.02001
REFERENCE PRESSURE (mm Hg) = 0.480
(N/sq. m.) = 0.5332E+02
SHAFT SPEED (rad/sec) = 149.0
(RPM) = 1423.
MODEL SUPPLY CURRENT (Amperes) = 50.42
HEAT LOAD/UNIT AREA (W/sq m) = 0.4312E+05

TEMPERATURE RATIOS:  T S. COOLANT (oK) = 330.7
T1= 1.251  T2= 0.973  T3= 0.976  T4= 1.000  T5 = 1.000
T6= 1.031  T7= 1.023  T8= 0.918  T9= 1.295  T10 = 1.333

PRESSURE RATIOS:  P S. COOLANT (N/sq.m.) = 0.196E+06
P1= 1.000  P2= 0.906  P3= 1.043  P4= 0.931  P5= 0.945  P6 = 0.000

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m.) = 0.138E-04
THERMAL CONDUCTIVITY k (W/m. oK) = 0.113E-01
Cp (J/kg oK) = 631.0
Cp/Cv = 1.1223
Pr = 0.7701

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE= 17.945  IMPINGEMENT JET= 57.088

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.2908E+05  Re Jet = 0.7473E+05
Ro " = 0.5854E+02  Ro " = 0.1862E+03
M " = 0.1123E+00  M " = 0.3588E+00

AVERAGE NUSSELT NUMBERS:
1= 0.500E+00  2= 0.000E+00  3= 0.427E+03  4= 0.310E+03
5= 0.247E+03  6= 0.205E+03  7= 0.176E+03  8= 0.000E+00
9= 0.000E+00  10= 0.482E+03  11= 0.375E+03  12= 0.278E+03
13= 0.225E+03  14= 0.183E+03  15= 0.264E+03
TEST # 60 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:  # OF JETS = 12  JET DIA (m) = 0.266E-02
SUPPLY PLENUM AREA (sq. m) = 0.130E-03
" PERIMETER (m) = 0.498E-01
ST. ANGLE = 30.00 HUB/TIP = 0.888 MEAN RADIUS = 0.457m
s/d = 2.0  p/d = 2.963  D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 20.0
FLOWMETER TEMPERATURE (OK) = 255.1
(OF) = -0.4
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00988
REFERENCE PRESSURE (mm Hg) = 0.480
(N/sq. m) = 0.5332E+02
SHAFT SPEED (rad/sec) = 153.1
(RPM) = 1462.
MODEL SUPPLY CURRENT (Amperes) = 46.38
HEAT LOAD/UNIT AREA (W/sq m) = 0.3602E+05

TEMPERATURE RATIOS:  T.S. COOLANT (OK) = 326.3
T1= 1.258  T2= 0.967  T3= 0.970  T4= 1.000  T5= 1.013
T6= 1.039  T7= 1.034  T8= 0.984  T9= 1.349  T10= 1.384

PRESSURE RATIOS:  P.S. COOLANT (N/sq.m.) = 0.117E-06
P1= 1.000  P2= 0.941  P3= 1.078  P4= 0.971  P5= 1.014  P6= 0.850

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:

VISCOSITY  u (Nsec/sq.m.) = 0.136E-04
THERMAL CONDUCTIVITY  k (W/m. OK) = 0.111E-01
Cp  (J/kg OK) = 627.1
Cp/Cv = 1.1231
Pr = 0.7719

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 14.634  IMPINGEMENT JET = 47.673

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.1454E+05  Re Jet = 3.3737E+05
Ro = 0.4664E+02  Ro = 0.1513E+03
M = 0.9219E-01  M = 0.3911E+00

AVERAGE NUSSELT NUMBERS:
1 = 0.000E+00  2 = 0.000E+00  3 = 0.244E+03  4 = 0.178E+03
5 = 0.142E+03  6 = 0.125E+03  7 = 0.115E+03  8 = 0.000E+00
9 = 0.000E+00  10 = 0.268E+03  11 = 0.210E+03  12 = 0.157E+03
13 = 0.134E+03  14 = 0.122E+03  15 = 0.156E+03
TEST # 61 TASK # 2:  MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:  # OF JETS = 12  JET DIA (m) = $0.286E-02$
           SUPPLY PLENUM AREA (sq. m.) = $0.138E-03$
           " " PERIMETER (m.) = $0.498E-01$
           ST. ANGLE= 30.0° HUB/TIP= 0.800 MEAN RADIUS= 0.457m
           s/d   = 2.0  p/d   = 2.963 D/d   = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 9.9
           (kg/sec) = 264.6
           (of) = 16.7
           CORRECTED R-12 MASS FLOW
           REFERENCE PRESSURE (mm Hg) = 0.350
           (N/sq. m.) = 0.466E+02
           SHAFT SPEED
           (rad/sec) = 154.0
           (RPM) = 1470
           MODEL SUPPLY CURRENT
           (Amperes) = 37.48
           HEAT LOAD/UNIT AREA
           (W/sq m) = 0.2383E+05

TEMPERATURE RATIOS:  T S. COOLANT (°K) = 329.4
                      T1= 1.225  T2= 0.940  T3= 0.942  T4= 1.000  T5 = 1.821
                      T6= 1.937  T7= 1.046  T8= 0.906  T9= 1.362  T10 = 1.399

PRESSURE RATIOS:  P S. COOLANT (N/sq. m.) = 0.633E+05
                   P1= 1.000  P2= 0.909  P3= 1.011  P4= 0.953  P5 = 0.949  P6 = 0.801

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
           VISCOSITY u (Nsec/sq.m.) = 0.138E-04
           THERMAL CONDUCTIVITY k (W/m. °K) = 0.112E-01
           Cp  (J/kg °K) = 629.8
           Cp/Cv = 1.125
           Pr = 0.7707

COOLANT VELOCITIES (m/sec):
           SUPPLY PASSAGE= 13.354  IMPINGEMENT JET= 43.892

NONDIMENSIONAL PARAMETERS:
           Re Supply = 0.766E+04  Re Jet = 0.181E+05
           Ro " = 0.421E+02  Ro " = 0.138E+03
           M " = 0.837E-01  M " = 0.275E+00

AVERAGE NUSSELT NUMBERS:
           1= 0.800E+00  2= 0.790E+00  3= 0.138E+03  4= 0.105E+03
           5= 0.800E+02  6= 0.756E+02  7= 0.747E+02  8= 0.800E+00
           9= 0.500E+00  10= 0.153E+03  11= 0.124E+03  12= 0.883E+02
           13= 0.818E+02  14= 0.790E+02  15= 0.912E+02
TEST # 52 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" PERIMETER (m.) = 0.498E-01
ST. ANGLE = 30.00 HUB/TIP = 0.800 MEAN RADIUS = 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 9.8
FLOWMETER TEMPERATURE (oK) = 268.2
(of) = 23.1
CORRECTED R-12 MASS FLOW (kg/sec) = 0.30475
REFERENCE PRESSURE (mm Hg) = 0.500
(N/sq. m.) = 0.6665E+02
SHAFT SPEED (rad/sec) = 209.8
(RPM) = 2003.
MODEL SUPPLY CURRENT (Amperes) = 37.61
HEAT LOAD/UNIT AREA (W/sq m) = 0.2399E+05

TEMPERATURE RATIOS: T S. COOLANT (oK) = 320.2
T1= 1.250 T2= 0.961 T3= 0.964 T4= 1.000 T5 = 1.029
T6= 1.047 T7= 1.075 T8= 0.915 T9= 1.304 T10= 1.417

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = 0.850E+05
P1= 1.000 P2= 1.053 P3= 1.197 P4= 1.053 P5= 1.232 P6= 0.201

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VIScosity = 0.134E-04
THERMAL CONDUCTIVITY k (W/m. oK) = 0.108E-01

Cp (J/kg oK) = 621.5
Cp/Cv = 1.1243
Pr = 0.7743

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 9.491 IMPINGEMENT JET = 31.890

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.7117E+04 Re Jet = 0.1829E+05
Ro " = 0.219E+02 Ro " = 0.7389E+02
M " = 0.6033E-01 M " = 0.283E+03

AVERAGE NUSSELT NUMBERS:
1= 0.000E+00 2= 0.000E+00 3= 0.134E+03 4= 0.101E+03
5= 0.764E+02 6= 0.716E+02 7= 0.781E+02 9= 0.800E+02
9= 0.000E+00 10= 0.146E+03 11= 0.118E+03 12= 0.331E+02
13= 0.769E+02 14= 0.822E+02 15= 0.881E+02
TEST # 63 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" PERIMETER (m.) = 0.498E-01
ST. ANGLE = 30.0° HUB/TIP = 0.800 MEAN RADIUS = 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 10.0
FLOWMETER TEMPERATURE (oK) = 267.5
(°F) = 21.8
CORRECTED R-12 MASS FLOW (kg/sec) = 0.00486
REFERENCE PRESSURE (mm Hg) = 0.600
(N/sq. m.) = 0.7998E+02
SHAFT SPEED (rad/sec) = 212.4
(RPM) = 2029
MODEL SUPPLY CURRENT (Amperes) = 31.55
HEAT LOAD/UNIT AREA (W/sq m) = 0.1688E+05

TEMPERATURE RATIOS: T S. COOLANT (oK) = 320.8
T1 = 1.185 T2 = 0.951 T3 = 0.952 T4 = 1.000 T5 = 1.023
T6 = 1.035 T7 = 1.056 T8 = 0.923 T9 = 1.311 T10 = 1.338

PRESSURE RATIOS: P S. COOLANT (N/sq. m.) = 0.119E+06
P1 = 1.000 P2 = 0.960 P3 = 1.024 P4 = 0.985 P5 = 1.034 P6 = 0.901

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY \( u \) (Nsec/sq.m.) = 0.134E-04
THERMAL CONDUCTIVITY \( k \) (W/m. oK) = 0.108E-01
\( \frac{C_p}{C_v} \) (J/kg oK) = 622.1

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 6.935 IMPINGEMENT JET = 23.571

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.7269E+04 Re Jet = 0.1868E+05
Ro " = 0.1587E+02 Ro " = 0.5393E+02
M " = 0.4404E-01 M " = 0.1498E+00

AVERAGE NUSSELT NUMBERS:
1= 0.000E+00 2= 0.000E+00 3= 0.147E+03 4= 0.105E+03
5= 0.827E+02 6= 0.753E+02 7= 0.839E+02 8= 0.000E+00
9= 0.000E+00 10= 0.162E+03 11= 0.123E+03 12= 0.902E+02
13= 0.815E+02 14= 0.908E+02 15= 0.940E+02
TEST # 64 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = \( \theta.206E-02 \)
SUPPLY PLENUM AREA (sq. m) = \( \theta.138E-03 \)
" PERIMETER (m) = \( \theta.498E-01 \)
ST. ANGLE= 30.00 HUB/TIP= 0.800 MEAN RADIUS = 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (\% of max) = 20.4
FLOWMETER TEMPERATURE (\(^\circ\)K) = 257.0
(\(^\circ\)F) = 3.0
CORRECTED R-12 MASS FLOW (kg/sec) = 0.01002
REFERENCE PRESSURE (mm Hg) = 0.650
(N/sq. m.) = 0.8664E+02
SHAFT SPEED (rad/sec) = 210.1
(RPM) = 2006.
MODEL SUPPLY CURRENT (Amperes) = 49.25
HEAT LOAD/UNIT AREA (W/sq m) = 0.4114E+05

TEMPERATURE RATIOS: T.S. COOLANT (\(^\circ\)K) = 337.1
T1= 1.284 T2= 0.945 T3= 0.949 T4= 1.000 T5 = 1.022
T6= 1.047 T7= 1.056 T8= 0.898 T9= 1.341 T10= 1.376

PRESSURE RATIOS: P.S. COOLANT (N/sq.m) = 0.149E+06
P1= 1.000 P2= 0.982 P3= 1.078 P4= 0.975 P5= 1.051 P6= 0.901

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY \( u \) (Nsec/sq.m.) = 0.141E-04
THERMAL CONDUCTIVITY \( k \) (W/m. \(^\circ\)K) = 0.116E-01
\( C_p \) (J/kg \(^\circ\)K) = 636.5
\( C_p/C_v \) = 1.1211
Pr = 0.7677

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 12.824 IMPINGEMENT JET = 39.850

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.1431E+05 Re Jet = 0.3679E+05
Ro " = 0.2782E+02 Ro " = 0.9218E+02
M " = 0.7468E-01 M " = 0.2477E+00

AVERAGE NUSSELT NUMBERS:
l= 0.080E+00 2= 0.090E+00 3= 0.279E+03 4= 0.198E+03
5= 0.155E+03 6= 0.132E+03 7= 0.142E+03 8= 0.008E+00
9= 0.008E+00 10= 0.309E+03 11= 0.234E+03 12= 0.170E+03
13= 0.142E+03 14= 0.157E+03 15= 0.173E+03
TEST # 65 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E+02
SUPPLY PLENUM AREA (sq. m.) = 0.132E+03
" " PERIMETER (m.) = 0.496E+01
ST. ANGLE = 30.00 HUB/TIP = 0.800 MEAN RADIUS = 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 20.6
FLOWMETER TEMPERATURE (oK) = 256.9
(OF) = 2.7
CORRECTED R-12 MASS FLOW (kg/sec) = 0.01013
REFERENCE PRESSURE (mm Hg) = 0.785
(N/sq. m.) = 0.931E+02
SHAFT SPEED (rad/sec) = 211.3
(RPM) = 2017.
MODEL SUPPLY CURRENT (Amperes) = 46.15
HEAT LOAD/UNIT AREA (W/sq m) = 0.3613E+05

TEMPERATURE RATIOS: T S. COOLANT (oK) = 338.3
T1 = 1.256 T2 = 0.938 T3 = 0.941 T4 = 1.000 T5 = 1.018
T6 = 1.042 T7 = 1.044 T8 = 0.906 T9 = 1.319 T10 = 1.341

PRESSURE RATIOS: P S. COOLANT (N/sq.m.) = 0.149E+06
P1 = 1.000 P2 = 0.974 P3 = 1.055 P4 = 0.978 P5 = 1.046 P6 = 0.981

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY (Nsec/sq.m.) = 0.141E-04
THERMAL CONDUCTIVITY k (W/m. oK) = 0.117E+01
Cp (J/kg oK) = 637.6
Pr = 1.1208

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE = 12.178 IMPINGEMENT JET = 40.327

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.144E+05 Re Jet = 0.3705E+05
Ro = 0.2802E+02 Ro = 0.9278E+02
M = 0.7542E-01 M = 0.2502E+00

AVERAGE NUSSELT NUMBERS:
1 = 0.000E+00 2 = 0.000E+00 3 = 0.299E+03 4 = 0.212E+03
5 = 0.161E+03 6 = 0.137E+03 7 = 0.144E+03 8 = 0.000E+00
9 = 0.318E+03 10 = 0.318E+03 11 = 0.240E+03 12 = 0.172E+03
13 = 0.144E+03 14 = 0.159E+03 15 = 0.182E+03
TEST # 66 TASK # 2: MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS: # OF JETS = 12 JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m) = 0.130E-03
" PERIMETER (m) = 0.498E-01
ST. ANGLE  = 30.00 DEG HUB/TIP = 0.800 MEAN RADIUS = 0.457m
s/d = 2.0 p/d = 2.963 D/d = 5.886

TEST DATA: VOLUME FLOW FROM FLOWMETER (% of max) = 41.0
FLOWMETER TEMPERATURE (oK) = 255.4
(oF) = 0.0
CORRECTED R-12 MASS FLOW (kg/sec) = 0.02020
REFERENCE PRESSURE (mm Hg) = 0.970
(N/sq. m) = 0.1293E+03
SHAFT SPEED (rad/sec) = 212.0
(RPM) = 2024.
MODEL SUPPLY CURRENT (Amperes) = 50.40
HEAT LOAD/UNIT AREA (W/sq m) = 0.4309E+05

TEMPERATURE RATIOS: T S. COOLANT (oK) = 334.9
T1= 1.258 T2= 0.959 T3= 0.961 T4= 1.000 T5= 1.013
T6= 1.003 T7= 1.029 T8= 0.927 T9= 1.283 T10= 1.301

PRESSURE RATIOS: P S. COOLANT (N/sq.m) = 0.245E+06
P1= 1.800 P2= 0.955 P3= 1.560 P4= 0.964 P5= 1.026 P6= 0.001

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY u (Nsec/sq.m) = 0.140E-04
THERMAL CONDUCTIVITY k (W/m. oK) = 0.115E-01
Cp (J/kg oK) = 634.6
Cp/Cv = 1.1215
Pr = 0.7685

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE= 14.616 IMPINGEMENT JET= 47.653

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.2902E+05 Re Jet = 0.7459E+05
Ro " = 0.3351E+02 Ro " = 0.1093E+03
M " = 0.9895E-01 M " = 0.2973E+00

AVERAGE NUSSELT NUMBERS:
1= 0.000E+00 2= 0.000E+00 3= 0.483E+03 4= 0.344E+03
5= 0.271E+03 6= 0.222E+03 7= 0.203E+03 8= 0.000E+00
9= 0.000E+00 10= 0.510E+03 11= 0.389E+03 12= 0.288E+03
13= 0.231E+03 14= 0.218E+03 15= 0.293E+03
TEST # 67 TASK # 2:  MEAN AND DIMENSIONLESS PARAMETERS

CONSTANTS:  
# OF JETS = 12  JET DIA (m) = 0.206E-02
SUPPLY PLENUM AREA (sq. m.) = 0.130E-03
" PERIMETER (m) = 0.498E-01
ST. ANGLE= 30.000 HUB/TIP = 0.800 MEAN RADIUS= 0.457m
s/d = 2.0  p/d = 2.963 D/d = 5.886

TEST DATA:  VOLUME FLOW FROM FLOWMETER (% of max) = 40.5
FLOWMETER TEMPERATURE (oC) = 254.8
(oF) = -1.0
CORRECTED R-12 MASS FLOW (kg/sec) = 0.01995
REFERENCE PRESSURE (mm Hg) = 1.250
(N/sq. m.) = 0.1666E+03
SHAFT SPEED (rad/sec) = 209.2
(RPM) = 1998.
MODEL SUPPLY CURRENT (Amperes) = 60.28
HEAT LOAD/UNIT AREA (W/sq m) = 0.6164E+05

TEMPERATURE RATIOS:  T S. COOLANT (oC) = 342.3
T1= 1.326  T2= 0.962  T3= 0.965 T4= 1.000 T5 = 1.016
T6= 1.046  T7= 1.041  T8= 0.916 T9= 1.306 T10= 1.338

PRESSURE RATIOS:  P S. COOLANT (N/sq.m.) = 0.231E+06
P1= 1.000 P2= 0.969 P3= 1.077 P4= 0.975 P5= 1.027 P6= 0.801

THERMOPHYSICAL CONSTANTS OF R-12 @ SUPPLY TEMPERATURE:
VISCOSITY \( u \) (Nsec/sq.m.) = 0.143E-04
THERMAL CONDUCTIVITY \( k \) (W/m. oK) = 0.119E-01
\( C_p \) (J/kg oK) = 641.0
\( C_p/C_v \) = 1.1201
Pr = 0.7657

COOLANT VELOCITIES (m/sec):
SUPPLY PASSAGE= 15.694  IMPINGEMENT JET= 50.826

NONDIMENSIONAL PARAMETERS:
Re Supply = 0.2889E+05  Re Jet = 0.7226E+05
Ro " = 0.3646E+02  Ro " = 0.1181E+03
M " = 0.9665E-01  M " = 0.3139E+03

AVERAGE NUSSELT NUMBERS:
1= 0.000E+50  2= 0.000E+50  3= 0.537E+03  4= 0.377E+03
5= 0.296E+03  6= 0.248E+03  7= 0.223E+03  8= 0.200E+02
9= 0.000E+00  10= 0.595E+03  11= 0.446E+03  12= 0.325E+03
13= 0.265E+03  14= 0.250E+03  15= 0.324E+03